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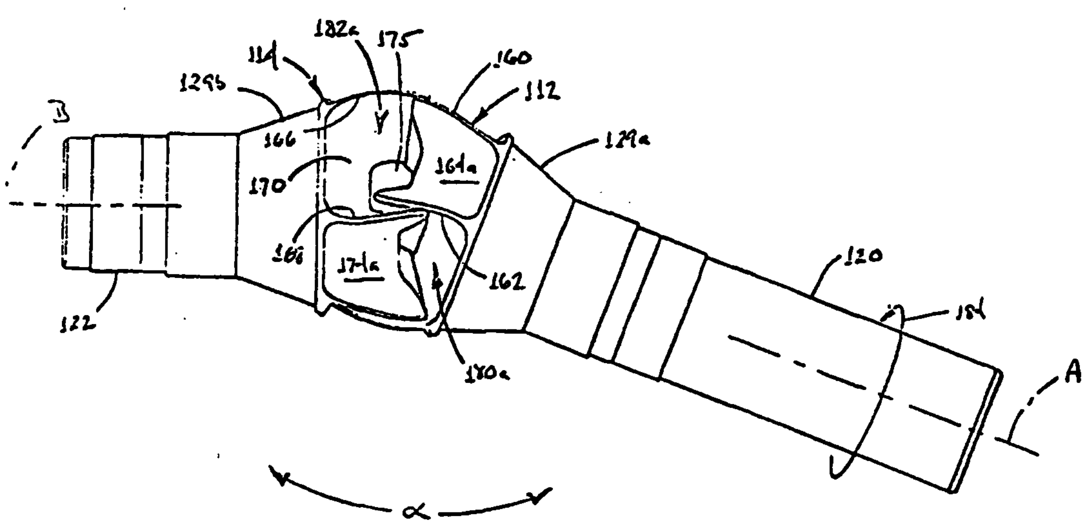


(57) Abstract

An improved rotary engine (10) and method for determining the contours of the sealing surfaces (24, 34) thereof. The improved engine provides for maintaining a predetermined, optimal gap (IG) between the sealing surfaces during rotation. The gap may be parallel or angled, and may be positive or negative so as to form an interference engagement. The rotors (112, 114) of the engine may be provided with mirror-image sealing surfaces (160, 162, 166, 168) so as to prevent development of excessive back-lash and clearance, and also to permit efficient reverse operation. The sealing surfaces (200, 212) may also be provided with recesses (208) for interrupting the seal at predetermined points in the rotational cycle, for enhanced wear characteristics and/or to accommodate abrasive or shear-sensitive fluids.

ROTARY ENGINE AND METHOD FOR DETERMINING ENGAGEMENT SURFACE CONTOURS THEREFOR

a. Field of the Invention

The present invention relates to rotary positive displacement engines and to methods for determining engagement surface contours for use in the making of rotary positive displacement engines.

b. Background

This invention concerns an advanced rotary positive displacement engine having high power to mass ratio and low production cost. The term "engine" as used in this patent document is taken to be a device that converts one form of energy into another. Hence, the term includes both devices which impart energy to the fluid flow (e.g. a pump)and those which employ the fluid flow to generate an energy output (e.g. an external combustion engine for providing a power source).

In the case of prior art combustion engines, the reciprocating piston type is most widely used for its low cost of production and efficient sealing, while the turbine has shown that an external combustion engine may offer greater power, partially from high speed. Rotary engines such as the Wankel engine have shown higher power-to-weight ratios than reciprocating engines but at the expense of increased fuel consumption. The present invention is a rotary device that offers many of the advantages of these prior art devices without many of their shortcomings.

In the case of pumps, there are many general types of pump designs known, such as positive displacement, centrifugal and impeller. Pumps of the positive displacement type are typically reciprocating or rotary. Many previous rotary combustion engine designs in turn, have

Low tolerance manufacturing techniques used for lower performance or less expensive designs will require a sealing surface geometry (SSG) which allows for the inconsistencies of the final surface. Higher tolerance machining techniques will also benefit from a predetermined SSG to maintain a minimum gap clearance or to prevent contact or binding of the mating rotors. Hard coating of a suitable base material also requires a pre coated surface geometry which prevents the coated SSG from binding or interfering

Some applications may even benefit from an interfering or "negative" SSG. Compressible or deformable materials and coatings can provide increased seal performance if they are designed to interfere with the mating surface on the opposite rotor. This can be accomplished by coating a harder material having a negative SSG to bring the surface back to a reduced negative SSG or a positive SSG.

Furthermore, fluid film bearings are used in industry to replace ball bearings or plain bearings in many applications. Fluid films for bearings range from several ten thousandths of an inch to several thousandths of an inch. Having a fluid film between the sealing surfaces of the engine rotors will decrease friction and wear, however, establishing this fluid film requires a correctly designed surface interface. If the surface interface has a gap space which does not account for the other variables which affect the fluid film, extra friction and wear, as well as volumetric efficiency compromises, may result.

An excessive clearance or gap between the sealing surface, however, may lead to excessive leak-by, thereby significantly impairing the overall efficiency of the engine. For example, if excessive "backlash" develops between the sealing surfaces of the CvR-type engine, this can result in undesirable amounts of leak-by.

Still further, it is desirable for many applications for the engine to be highly efficient in both forward and reverse directions of operation. Consequently, if the

each rotor being at the center of the cavity. The rotors interlock with each other to define chambers. Vanes defined by a contact face on one side of the vane and a side face on the other side of the vane protrude from the rotors. The contact faces of the rotors are defined so that there is constant linear contact between opposing vanes on the two rotors as they rotate. The side faces are preferably concave and extend from an inner end of one contact fact to the outer end of an adjacent contact face, equivalent to the tip of a vane. The side faces and contact faces define walls of chambers that change volume as the rotors rotate. Ports for intake and exhaust are preferably configured to have shapes complementary to the intersecting vanes of the rotors.

Also in accordance with the present invention, a method is provided for determining a precise, controllable gap between the sealing surfaces on the rotors. These methods include both mathematical and geometric processes, a well as methods for verifying that the correct contours have been imparted to the surfaces.

Still further, in accordance with a preferred embodiment of the invention, the vanes on the rotors are provided with mirror-image contoured sealing surfaces which both maintain the desired gap during operation by reducing back-lash, and which also permit efficient reverse operation of the engine.

These and other aspects of the invention will be described in more detail in what follows and claimed in the claims appearing at the end of this document.

contact surface contours in accordance with the present invention;

FIGS. 9A-9D are a series views of a visual model illustrating the method by which the contours of the contact surfaces are determined in the present invention, by conceptual rotation of predetermined system axes based on a predetermined mathematical relationship;

FIGS. 10A-10D are a series of computer-generated graphical images, illustrating the manner in which the contours of the contact surfaces are determined using the mathematical relationship in accordance with the present invention;

FIG. 10E is a perspective view of one of the rotors in accordance with the present invention, with the dotted line image showing the area of the contact surface having the contour which is generated through the steps shown in FIGS. 10A-10D.

FIG. 11A is a geometric figure, similar to FIG. 8C, showing a revised calculation of the contact surface contours to provide a modified tip-radius form having a slightly flattened shape for enhanced wear characteristics;

FIG. 11B is a partial, cross-sectional view of the tip portion of a contact surface contour formed in accordance with the relationship shown in FIG. 11A;

FIG. 12 is a schematic view showing the relationship of mirrored contact surfaces, somewhat similar to those shown in FIGS. 7A-7C, with these being configured to provide a predetermined spacing between adjacent contact surfaces so as to provide a predetermined fluid film thickness during operation and also to permit reverse operation of the engine;

FIG. 13A is a partial, enlarged view of adjacent tip portions of the mirrored contact surfaces of FIG. 12, showing the spacing between the tip surfaces in greater detail;

FIGS. 19A - 19C are a series of perspective, somewhat schematic views of a rotor assembly in accordance with an embodiment of the present invention in which relief areas are formed in the sides of the sealing surfaces between the upper and lower ends thereof so as to reduce wear and provide enhanced characteristics for certain applications; and

FIG. 20 is a chart demonstrating the relationship between the relative sliding velocity of the sealing surfaces of an engine in accordance with the present invention, as a function of shaft velocity.

within the housing in an axial direction, each being predominantly on one side of the common center of the rotors.

The portion of the interior surface 14 that is spherical is the portion in which both the vanes of the master rotor 20 and salve rotor 30 rotate. In an extreme position, where the vanes of one rotor extend into the shaft of the other rotor the vanes of both rotors extend into the shafts 22, 32. The shafts 22, 32 are not spherical, but rotationally symmetric. In addition, the master rotor 20 and slave rotor 30 should be generally spherical in the portions in which they overlap during operation. The remainder of the rotors 20, 30 and the interior surface 14 need only have rotational symmetry to the extent required to have the rotors 20, 30 rotate in the housing 12.

As will be seen, the contoured faces 24, 26, 34, 36 of the mater rotor 20 and slave rotor 30 cooperate with each other and the interior surface 14 of the housing 12 to form chambers 40 (the space between the faces of the rotors) that change volume with rotation of the rotors 20, 30 about the axes A and B respectively. Ports 42 are provided in the housing 12 to allow fluid flow in and out of the chambers.

Each contoured face is formed of a contact face 24, 34 and a side face 26, 36 defining vanes (blades) 25, 35 between them. The contact faces 24, 34 form areas of contact between the two rotors 20, 30. Sealing of the chambers 40 is accomplished by close tolerance fit of the rotors 20, 30 against the housing 12 and bearing 16, as well as the relationship of the vanes 25, 35 with respective contact faces 24, 34. As is described in the above-referenced US 5,755,196, the contours of the surfaces in a CvR engine of this type can be determined by defining the contact faces of the rotors by a locus which is formed as the rotors rotate about their respective axes by points on the other rotor, the points of each rotor that define the locus lying along an outer edge of a cone whose central axis

The material of the rotors housing the bearing 16 is in fact concave over greater than 180°, creating difficulties in construction. The bearing may be made integral with or otherwise fixed to either rotor, preferably the master rotor 20. For the other rotor, the bearing 16 can be loosely fitted in a less than 180 bearing housing, resulting in a greater leakage path, or the bearing may be press fitted into the housing, thermally contracted and inserted into the bearing housing, or slotted for insertion and rotated once inside the bearing housing to present a round bearing surface to the slave rotor.

As is shown in FIG. 1, the master rotor 20 is driven by a power source (not shown) through shaft 22. Vanes 25 of rotor 20 push on contact faces 34 of rotor 30 on the side shown on the other side (not shown) contact face 24 of rotor 20 push on vanes 35 of rotor 30.

The internal and external configuration of the housing is shown in FIGS. 2, 3 and 4. In particular, the location of the ports 42 can be clearly seen, along with flanges 50 for connection of the housing 12 to input and output pipes (not shown). An alternative threaded coupling 51 is also shown in FIG. 1. The housing 12 is preferably formed of two halves 12a and 12b bolted together with bolts 54. The ports 42 are located at opposed sides of the housing, with an intake port 42a and outlet port 42b. Areas 55 show contact areas of vane on contact faces between the master and slave rotors 20, 30. Fluid enters the intake port 42a in expanding chamber 40a. Chamber 40c is at maximum expansion in this rotational position. Chamber 40b is contracting and therefore forces fluid out of port 42b. Chamber 40d is at maximum compression in this rotational position. Preferably, the ports 42 have peripheries that match the chamber configurations at the point the chambers cross the boundaries of the ports so that as many points as possible of the chamber edge, defined by a pair of vanes 24, 34, cross the port edges at the same time. The trailing edge of

b. Mirrored Contact Surfaces.

A preferred embodiment of the invention is shown in FIGS. 5-7C, the engine in this exemplary embodiment being configured for use as a pump, although again it will be understood that the engine can be configured as an external combustion engine or other power source. As will be described in greater detail below, a particular enhancement featured in the embodiment which is shown in FIGS. 5-7c lies in the mirrored contact surfaces which are provided on the leading and trailing sides of the "lobes".

As with the embodiment of FIG. 1, the engine 100 as shown in FIG. 5 includes a master or power rotor 112 which rotates about a first axis A and slave or passive rotor 114 which rotates about a second axis B which is offset from the axis A by an angle θ (see FIG. 7A). The rotors are housed between the two halves 116a, 116b of an external casing which seals and supports the assembly and also has inlet and outlet ports for the flow of fluid through the engine.

Each rotor 112, 114 is partially spherical with a common center, and the casing includes a corresponding spherical cavity 118 which receives and holds the rotors in engagement. The end shafts 120, 122 of the master and slave rotors are supported by the casing. The end 124 of the latter terminates and is fully enclosed within the casing 116, which provides the advantages of simplified sealing and reduced cost of manufacture, although it will be understood that in some embodiments the slave rotor shaft may extend through the exterior of the casing. The master rotor end shaft 120, in turn, extends outwardly from the casing and is connected to a suitable external power source (not shown), such as an electric, hydraulic or other motor.

Each end shaft is supported in a pair of bearings 126 and 128 to maintain shaft stability and eliminate end play. The inner bearings 126 include conical bearing faces (not

4, however, each vane or lobe is provided with two mirror image contact surfaces, i.e., a leading contact surface and a mirror image trailing contact surface.

The relationship between the leading and trailing contact surfaces is perhaps best seen in FIG. 7B, which is the top or "overhead" view of the master and slave rotors 112, 114. As can be seen, the lobes 164a, 164b, etc. of the master rotor 112 are angularly spaced so as to define a plurality of angularly spaced cavities 172, and the lobes 174 on the slave rotor define corresponding cavities 176. As can also be seen, each lobe is received in the corresponding cavity in the opposite rotor i.e., the master rotor lobes 164 are received in the cavities 176 in the slave rotor, and the slave rotor lobes 174 are received in the cavities 172 in the master rotor. The area in the center of the rotors, between the lobes on either side, is sealed by a ball 175 or other generally spherical body.

As also can be seen in FIG. 7B, the leading and trailing contact surfaces on each lobe engage the corresponding contact surfaces on each socket (these being the contact surfaces of the lobes on either side of the socket), as indicated at the areas 178. Consequently, a series of sealed chambers 180a, 180b, 180c are formed about the end of the master rotor, between the ends or "heads" of the lobes in the bottom of the cavities, and a corresponding series of sealed chambers 182a, 182b, 182c are thus formed around the end of the slave rotor.

The chambers change in volume with rotation of the rotor assembly, in the direction indicated by arrow 184. As can be seen by comparison of chambers 180a and 182a in FIG. 7A, the volume of the chamber increases as these rotate past the inlet port 142 (see FIG. 5), thus drawing fluid into the pump. The ports are shaped so that each chamber moves out of register with the inlet port just as the chamber reaches its maximum volume (see chamber 180b in FIB. 7B), and just before the chamber begins to rotate into register with the

depending on operating speed, back pressure, fluid viscosity and other factors, an equilibrium level is achieved in which a fluid film exists between both leading and trailing surfaces.

Additional advantages include increased strength of the rotor lobes, since the area between the mirrored contact surface (i.e., the backs of the contact surfaces) can be filled in, so that the back side of each of the faces is reinforcing the other, giving the lobes strength comparable to that of a gear tooth. Also, because of the higher strength, it is possible to operate the pump at higher pressures, which is advantageous in increasing the power ratio, or power density, of the pump.

b. Mathematical Calculation of Contact Surface Contours

The manner in which the contours of the contact surfaces are determined mathematically will now be described with reference to FIGS 8A-10D.

FIGS. 8A-8D provide a series of graphical representations of axes, vectors, angles, and other values associated with the mathematical computation of the contact surfaces of the vanes/lobes, as follows:

FIG. 8A shows the orientation of the two rotor axes, Axis 1 and Axis 2, intersecting at O and placed at an angle A° apart. The line O-O is initially in the plane of the two axes and bisects the direction of each, so that it makes an angle of (90+A/2)° with each axis direction. Point Q is a radial line on the surface of a sphere o radius R, which is a point locating the working surface of the rotor attached to shaft [2]. The plane P formed by the line O-O and OQ will be a plane that changes orientation in space.

When the pump turns about each axis by the same angle, $\theta = {}^{\theta}1 = {}^{\theta}2$ as shown. To construct the rotor surface on Axis 2, we need to consider the relative motion of Axis 1 with

from the vector cross product to get the velocity of Q.

$$V_Q = u0 \times Rou1$$
$$V_Q = [\cos\frac{\Delta}{2}\cos\theta,\quad \cos\frac{\Delta}{2}\sin\theta, -\sin\frac{\Delta}{2}] \times [x, y, z]$$

where the component of the two vectors are given. The vector uO is a unit vector with direction along O-O, which changes with rotation, as does u1.

The unit vector along the tangent of the path of Q' or Q will be

$$u2 = \frac{V_Q}{|V_Q|}$$

and will be a function of both the shaft angle A and the rotation θ.

A third unit vector, perpendicular to both u1 and u2 will be

$$u3 = u1 \times u2$$

and will be in a transverse direction, along QQ'.

The coordinates of the point Q can now be determined from the vector equation,

$$OQ = R = R\cos\delta\ u1 + R\sin\delta\ u3$$

in which

$$OQ' = R_O - R\cos\delta\ u1$$

The outer edge of the surface determined by Q is shown in FIG. 8D. Also shown is the rotation of plane P for different rotations of the shafts.

The total angular twist S, along the axis O-O in any general position can be most easily obtained by determining the angular change in the normal to the plane P, which is the unit vector u2. This vector is always directed along the tangent to the path of Q or Q', and has already been defined.

As is shown in FIG. 8E, the untwisted position of the plane P can be obtained by rotating the plane and its initial normal direction vector u2 about the z axis in the xy plane through the angle θ, to a new position $u2'_O$ and

following the mathematical process set forth above, the point Q sequentially plots out a line having a contour of a line on the contact surface of the lobe.

For purposes of illustration, FIGS. 9A-9D show rotation of Axis 1 90° from the start position to the final position; it will be understood, however, that determination of the line is ordinarily carried out in small degree increments, so as to define a smooth, continuous contour.

Accordingly, FIG. 9B shows the model 190 with the Axes 1 and 2 having been rotated together by an angle θ of 90°, so that axis R swings from the vertical alignment (for purposes of illustration) shown in FIG. 9A to the horizontal alignment in FIG. 9B. Then, with Axis 1 held stationary, Axis 2 is rotated back by an angle $-\theta$, which is equal to θ but in the reverse direction, rotating axis R to the position which is shown in FIG. 9C. Finally, the axis R is rotated by the amount θ_S which is calculated in accordance with the mathematical system described above, bringing point Q to its final position Q", as shown in FIG. 9D. For purposes of illustration, FIG. 9D also includes a broken-line image 192 which shows the original position of point Q at the start point shown in FIG. 9A.

FIGS. 10A-10D are a series of views similar conceptually to FIGS. 9A-9D, but showing the manner in which the above process is used to generate or determine a contoured line 194 in a computer plotting program. As can be seen, by following the process described above, the point Q is moved sequentially from position to position line 194, with each rotation of the Axes 1 and 2. By in essence "connecting the dots", i.e., the position of point Q at each position of Axis 1, a continuous contour line is created which corresponds to the contour line along one of the contact surfaces, such as the contact surface 160 on lobe 164, as shown in FIG. 10E.

The offset establishes sufficient clearance between the contact surfaces to establish the fluid film and avoid the

The offset distance from the axis O-O out to the working surface is (4 sinδ + t), where s = Rδ. If the tip were to be reshaped to provide a larger radius of curvature at the beginning of contact (for the purpose of reducing wear), the profile of the working surface can still be calculated readily from the existing computer program.

An approach is as follows, as shown in FIGS. 11A and 11B. Modify the tip radius to make a slightly flattened shape, in the vicinity of where first contact occurs. This shape can be identified as $s=s(\theta)$, which means the radius is a function of (depends on) the shaft rotation θ. Once this is selected, the radius can be input as a function of small angular increments, and the profile of the mating working surface calculated for the same fluid thickness t. Actually, fluid thickness may not be constant everywhere. It will probably depend on the relative sliding velocity of the vanes, which increases from zero at the point of contact and increases to a maximum near 90° rotation. The initial flattening of the tip may affect this also.

The working surface would normally follow a radial line towards the center O, resulting in a film thickness that tapers towards the center. The relative sliding velocity between adjacent lobes will be highest at the outside, so a larger thickness of film there seems reasonable. However, for applications where small particulates are contained in the fluid, it may be better to machine the rotor so that a parallel gap is produced. This may prevent material from sticking in the small end of the tapered gap, even though it would tend to be flushed away during the next rotation.

In FIGS. 12, 13A, and 13B, c is the distance between centers of adjacent tips (measured along the arc of the surface of a sphere of radius R). The arc length C is the distance between like lobe shapes (circular pitch length).

$$C=2(c=2s+t) = \frac{2\pi R}{n}$$

FIGS. 13-19 illustrate a method for geometric determination of the contact surface contours consistent with the mathematical calculations described above, but which corresponds more directly to an actual manufacturing process for forming the surfaces, as by hobbing material from a blank so as to form the lobes and surfaces.

Two of the main considerations when determining the correct sealing surface gap (SSG) are the "lift off clearance" and the contact characteristic. The "lift off clearance" is the thickness of the fluid film between the sealing surfaces of the two rotors when the engine is operating in its intended mode. "Lift off clearance" is affected by the speed of the engine, the viscosity of the fluid medium, and the differential pressure between the inlet port and the discharge port. Contact happens when the one or two or all of these factors is insufficient to maintain a fluid film between the mating surfaces.

The contact characteristic describes how the sealing surfaces mate when the fluid film is not sufficient to achieve "lift off". The three basic types of contact are (1) Full radial contact. (2) Inner radial contact. (3) Outer radial contact. These characteristics can be different at different angles of rotor rotation.

Maintaining a fluid film is desirable to reduce wear, as well as to allow entrained particles to pass between the sealing surfaces without damaging the particles or causing excessive abrasion to the sealing surface.

Some design considerations which should be taken into account to achieve a fluid film during engine operation are as follows:

U.S. 5,755,196 describes a CvR engine configuration with a "contact" or "close tolerance" seal design which does not optimize or account for the "lift off situation". This type of surface geometry relies on a line to line seal between the rotors and is intended to operate with each

produce rotor seal surfaces which interfere more towards the center of the rotors than they do toward the outside of the rotors. The advantage of this design would include better sealing near the center of the pump, and lower friction and less resistance further from center where any resistance will have a greater effect on the operating efficiency of the engine.

To achieve this "angular interface" effect, as well as the "parallel interface" effect, it may also be necessary to introduce a second surface shape variable, which is the angular position of each contact face about the center axis of the pump rotor. By rotating each seal surface relative to the rest of the pump, a predetermined surface interface with specific characteristics can be achieved.

The angular position effect, and the off-center cone apex effect will be covered in the following description of how to achieve the desired sealing surface geometry:

Referring now to FIG. 4, a spherical rotor RA is positioned for rotation about its center axis AA. A second axis AB is positioned at an angle X to axis AA. A cone C is positioned with its center axis collinear with a line Y that bisects the obtuse angle between axis AA and axis AB.

If a positive parallel SSG is desired, the cone C is positioned on line Y with its apex X below the point P where the two rotor axes intersect.

If a negative parallel SSG is desired, the apex of the cone must be positioned above the point P. (The smaller the angle of the cone, the more its apex must be positioned off center to achieve a given gap clearance or interference.)

As is shown in FIGS. 15A-E, the spherical rotor and the cone are then rotated around their respective axes (i.e., cone C rotates on axis AB at a fixed angle thereto) and the path of the cone is removed from the spherical rotor. This will define the "seal surface" S of one side of one vane V_1 on the rotor RA.

The rotor is then rotated toward the first cone and another cone shape C is positioned with its axis collinear with the line Y. This cone has the same angle as the first cone and it is positioned with its apex the same distance from center but on the opposite side of point P (see FIG. 15). This cone is added to the rotor RA and becomes the "seal tip" T of this seal face, as is shown in FIG. 15E. The sequence is then repeated for the second rotor RB (See FIGS.16A-16B) with a cone which is positioned along the center axis of the adjacent "seal tip" T cone of the rotor RA.

Once this sequence is repeated for each side of each vane, the engine will have a predetermined parallel interface gap IG between mating surfaces as is shown most clearly in FIG. 16B.

The other gap configuration which can be used on its own or in combination with the "offset cone" gap configuration, is the "angular interfacial gap".

This type of gap (or interference) is achieved by rotating each seal surface around the center of its rotor's axis relative to the seal surface on the opposite side of the vane it is on as is shown in FIG. 17. Comparative examples of positive and negative "angular" and "parallel" interfacial gaps are shown in FIG. 18.

An angular interfacial gap may offer performance benefits for certain applications. For example, the centrifugal force of the rotation of the engine could be used to force particulate matter entrained in the fluid to the periphery of the engine chamber. In this case an angular interfacial gap with a larger gap at the periphery of the rotors would allow the particles to pass through the thicker fluid film, while a more efficient seal could be maintained closer to the center of the rotors where the fluid film is thinner.

A characteristic of the "parallel interfacial gap" compared to the "angular interfacial gap" is that the

interfacial gap at an angle in opposition to the initial angular interfacial gap.

Some transitional gaps will be a variation of the above description in that they will incorporate only one or two of these descriptions.

Although the cone shape described above is the ideal shape, and the simplest to calculate and design, it will be understood that other similar shapes (such as a portion of a much larger cone or simply a sharp edge) could be used, however, as the mating surface is designed to maintain the desired SSG as both rotors spin at the same speed.

Furthermore, it will be understood that, while the description of the method of the present invention has been described herein with regard to externally contoured vanes/lobes, the method is equally applicable to CvR engines having pistons or corresponding structures which are house or retained within the lobes, such as the piston-engine structure which is shown in FIG. 16 of the above-referenced U.S. patent.

e. Verification of Contours

Many methods for verifying the surface shape are available. A contact CMM machine, for example, could be used to determine a number of points on the surface of a completed rotor, and establish what the seal surface characteristic is. The most basic way of determining if a rotor design has been manufactured according to the present invention is to create a plane which is perpendicular to a point on the seal face (or seal tip) which passes through the spherical center of the sphere. Two points on the seal face or seal tip surface which are also on this plane will be connected and extended toward the spherical center of the engine.

portions of the rotation of the assembly, i.e., at those points during the rotation where the seal is required in order to maintain efficiency. This configuration is advantageous in a number of applications, including for use with pumping sheer sensitive or abrasive fluids, and for enhanced wear characteristics.

Accordingly, as can be seen, in the embodiment which is illustrated in FIGS. 19A-19C, the sealing surfaces 200 on the vanes 202 of the two rotors 204, 206 are each formed with a recess or channel area 208 which extends radially across the rotor base and separates the sealing surface segments 210, 212 which lie proximate the tip and at base portions of the contoured face.

The sealing surface segments 210, 212 are formed in accordance with the methods described above, i.e., these are configured to form the requisite seal with the corresponding segments on the adjoining contoured face, with a predetermined gap as desired. Since the sealing segments are formed at the top and bottom of each surface, the rotors form an effective seal only when the chambers defined thereby are approximately at top and bottom dead center, as is shown in FIGS. 19A and 19B.

At points in the cycle between top and bottom dead center, however, the channels 208 eliminate direct contact between the two sealing surfaces so as to form a relief gap 220, as is shown in FIG. 19C. The relief gap reduces sheer stresses on fluid in this area, and also allows particulate or abrasive material to pass therethrough without causing wear against the sealing surfaces. Furthermore, the relief gap reduces wear by eliminating a potential content between the sealing surfaces during the intermediate phases of the engine cycle, even in applications not being used with abrasive fluids. Since sealing is only critical when the chambers are at top and bottom dead center, these advantages are achieved without significant cost to the overall efficiency of the engine.

WHAT IS CLAIMED IS:

1. An engine, comprising:

a housing;

a first rotor mounted for rotation on the housing about a first axis, said first rotor including first and second opposite facing contoured faces and defining at least part of a sphere having a center;

a second rotor mounted for rotation on said housing about a second axis said second rotor including third and fourth contoured faces and defining at least part of a sphere having a common center with said center of said first rotor;

said first and second and said third and fourth contoured faces being mirror image identical and being arranged in face-to-face engagement;

so that said engagement of said mirror-image contoured faces prevents backlash between said rotors so as to maintain a predetermined gap between said faces during operation of said engine.

2. A method for determining a contoured contact faces for a rotor of an engine so as to provide a predetermined gap between said faces, said method comprising the steps of:

providing a first rotor for being mounted on a housing for rotation about a first axis in engagement with a second rotor which is mounted on said housing for rotation about a second axis which is offset from being collinear by an angle α and which intersects said first axis at a common center of said rotors;

defining a cone which is in contact with said first rotor and which has an apex and an axis which bisects the obtuse angle between said first and second axes of said rotors;

angled so as to increase in width radially inwards towards said common center of said rotors.

8. An engine, comprising:

first and second rotors mounted for rotation on a housing, said rotors having contoured surfaces which engage to form chambers during said rotation thereof;

each said contoured surface comprising:

an upper edge area and a lower edge area for engaging upper and lower edge areas of an adjacent contoured surface so a to seal said chambers at predetermined points during rotation of said first and second rotors;

said upper and lower edge areas of said contoured surfaces being separated by a recessed zone between said edge areas which forms a gap between said surfaces at predetermined points during rotation of said first and second rotors so as to permit passage of fluid therethrough.

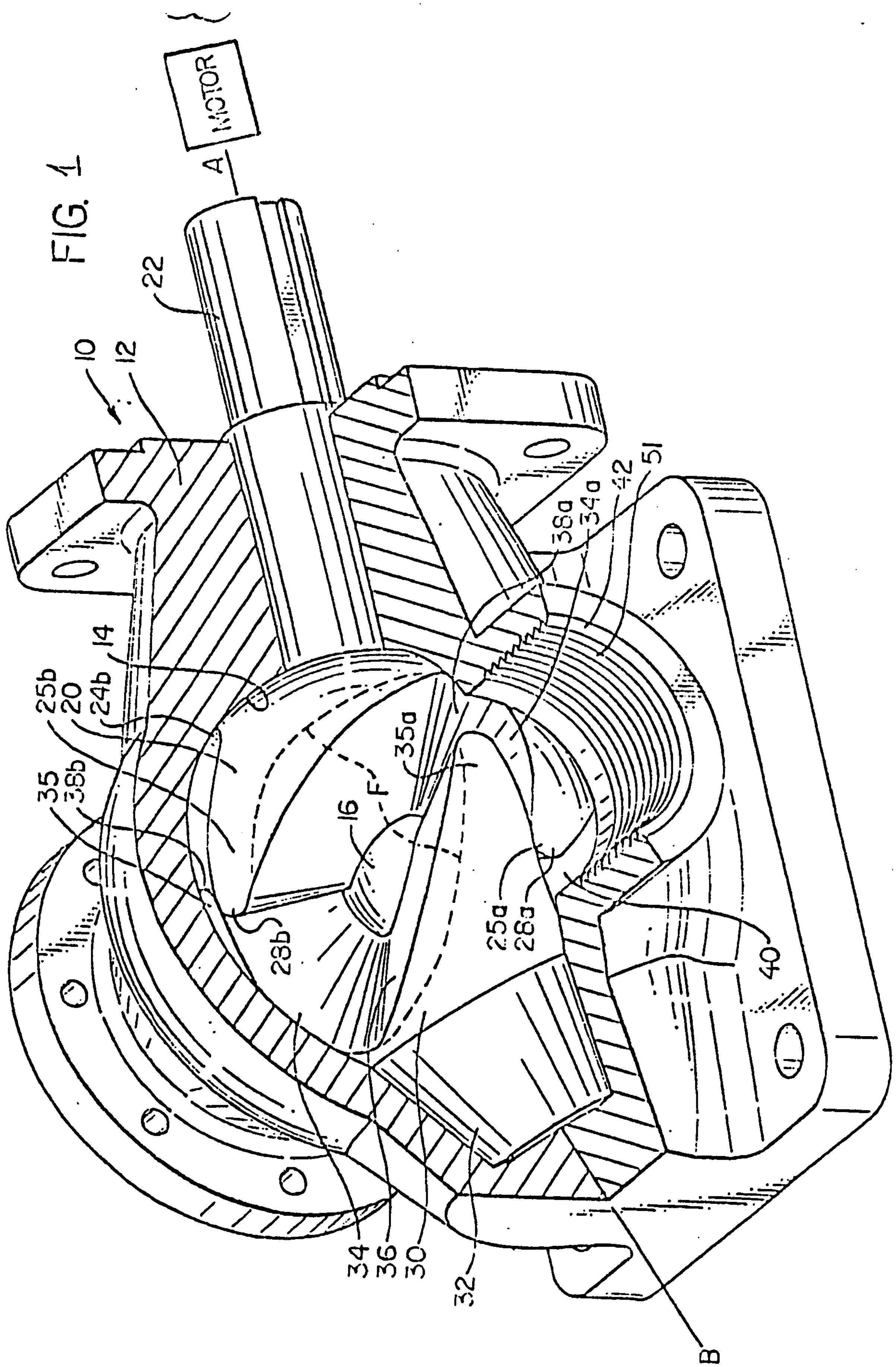

MOTOR
A
22
10
12
14
20
24b
25b
35
38b
16
F
35a
38a
34a
42
51
28b
25a
28a
40
34
36
30
32
B

FIG. 1

FIG. 2

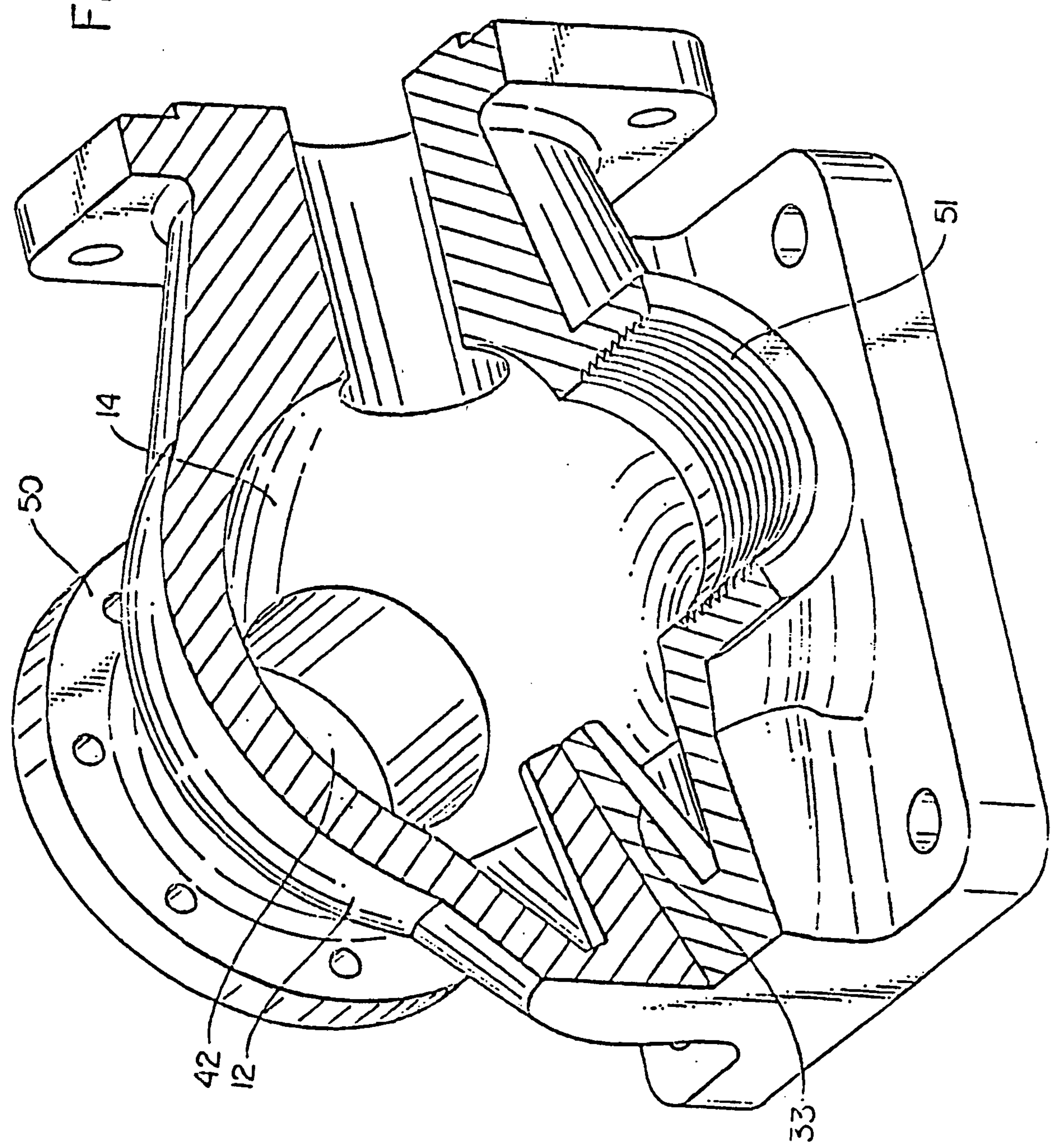


FIG. 3

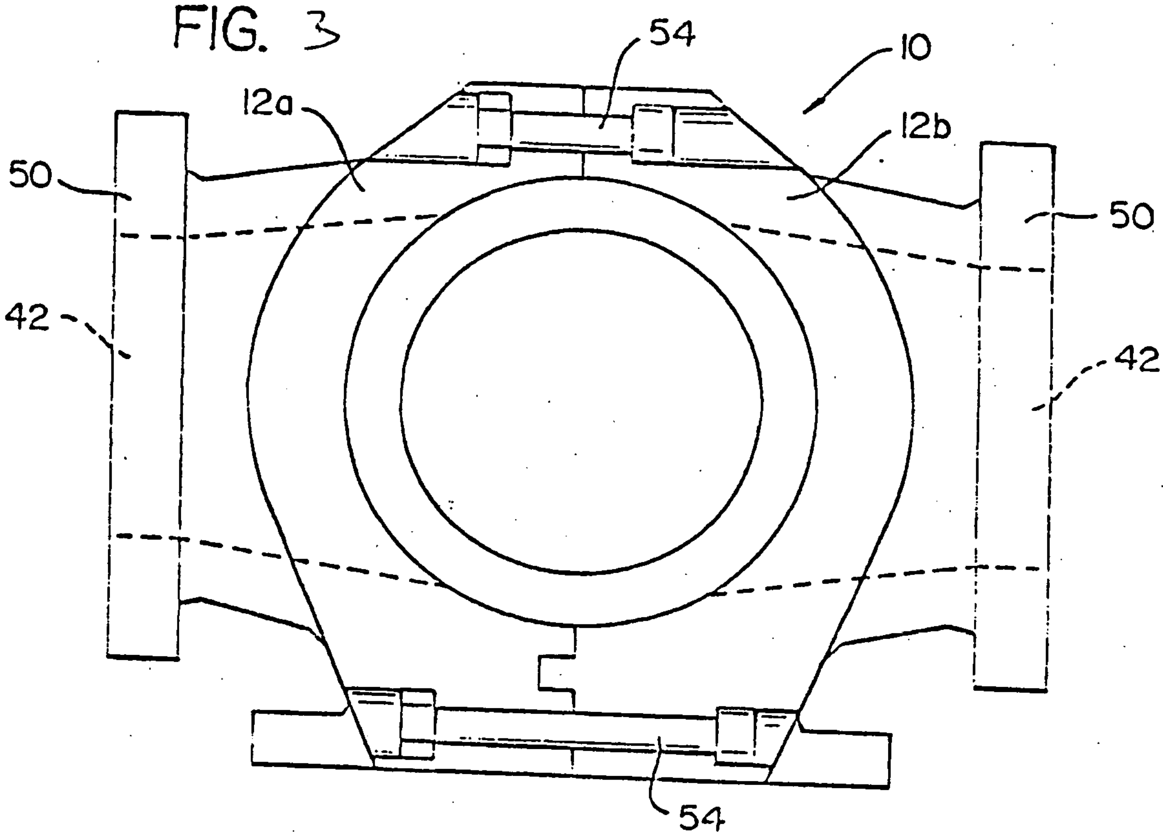
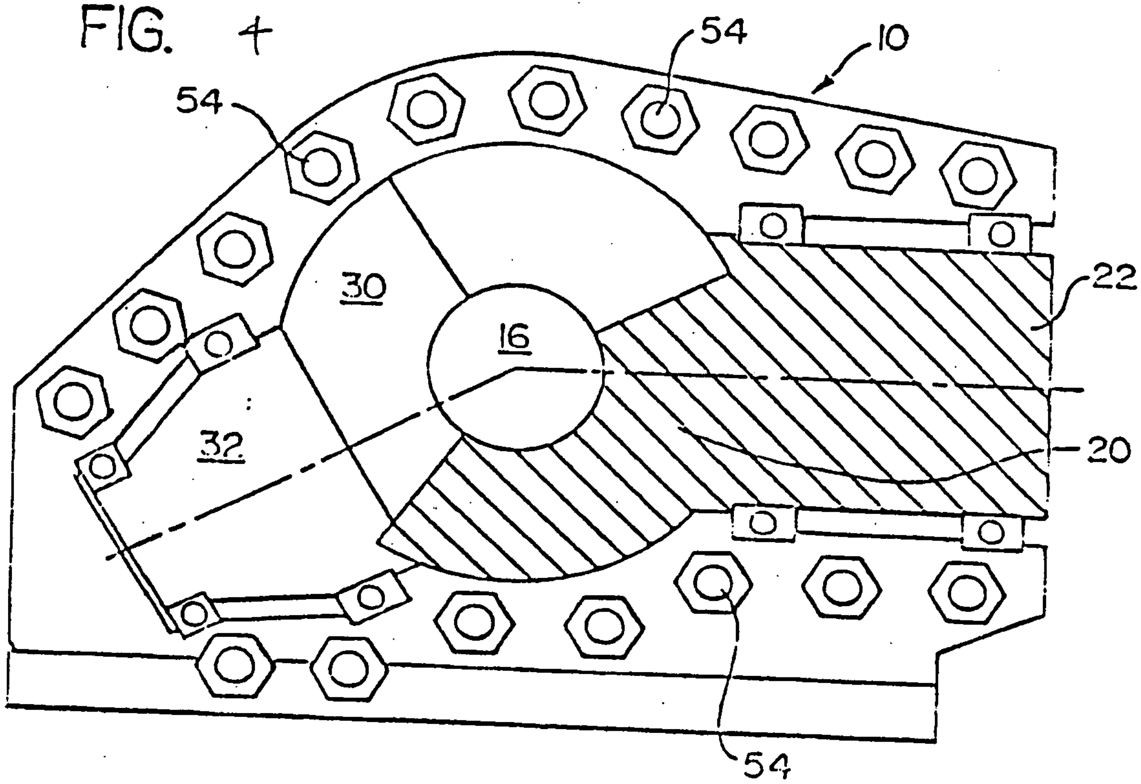


FIG. 4



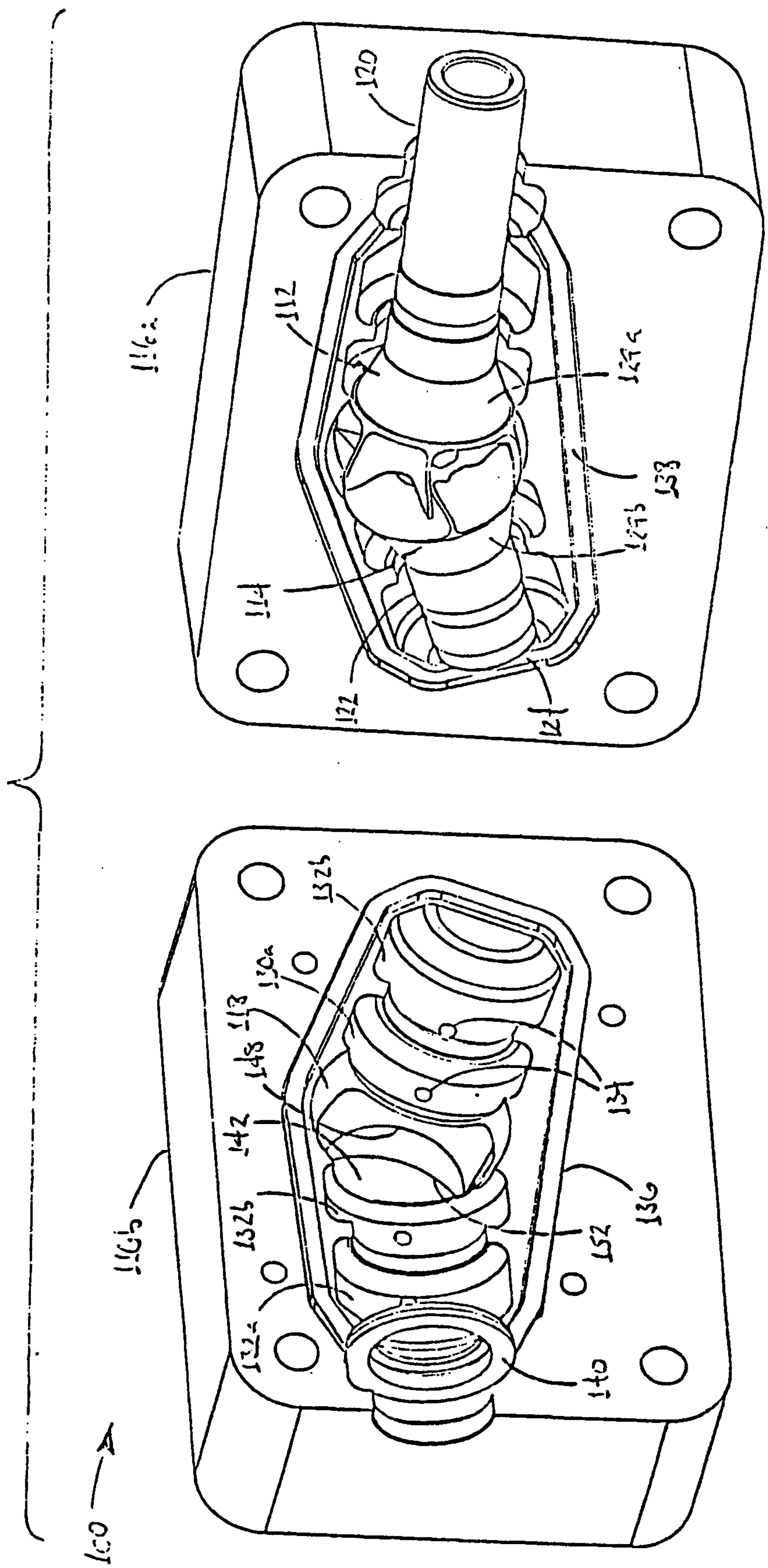

120
112
116a
124a
133
124b
114
122
121
116b
132b
130a
148
113
142
134
136
152
132a
140
100

FIG. 5

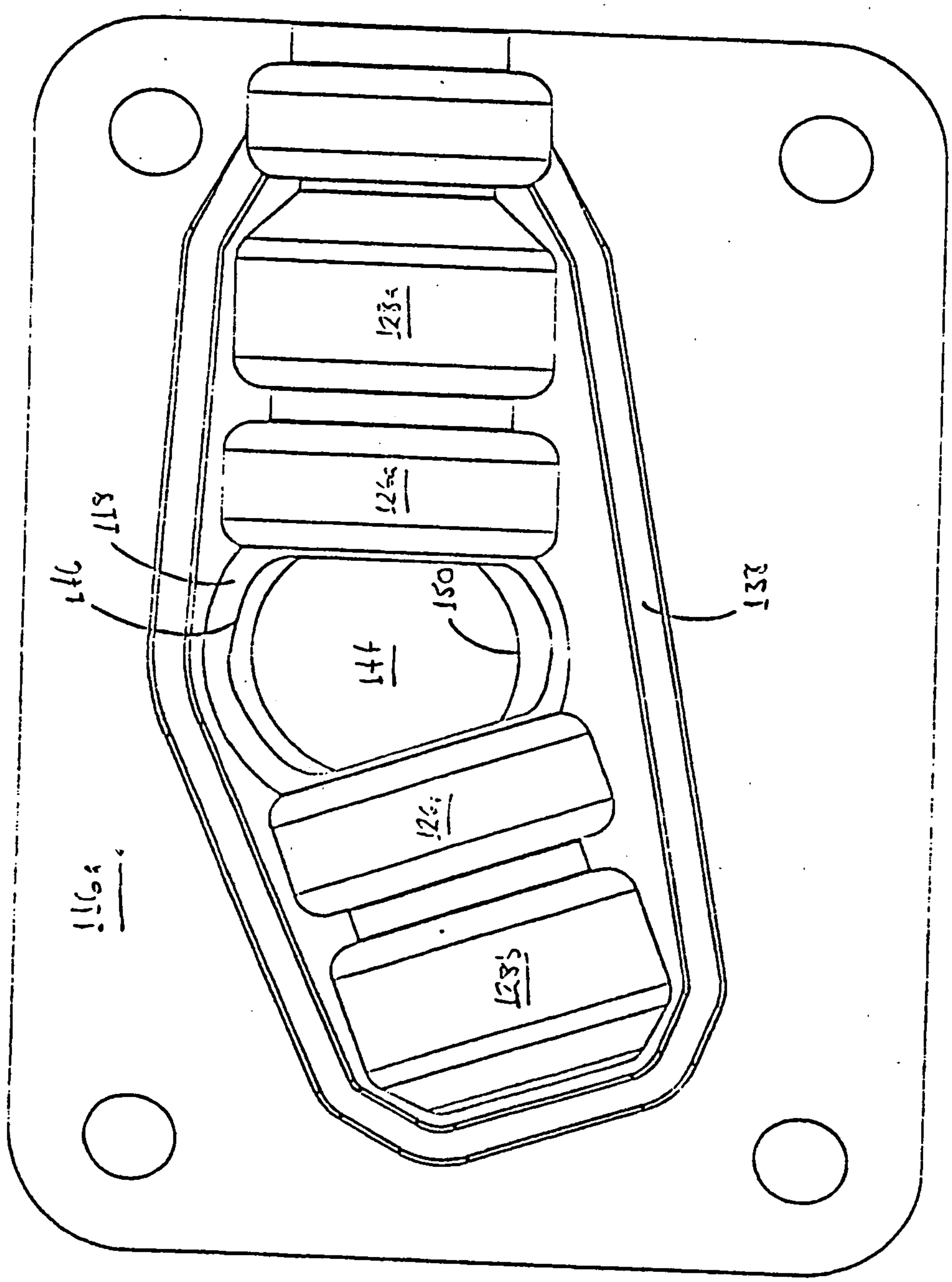


FIG. 6

FIG. 7H

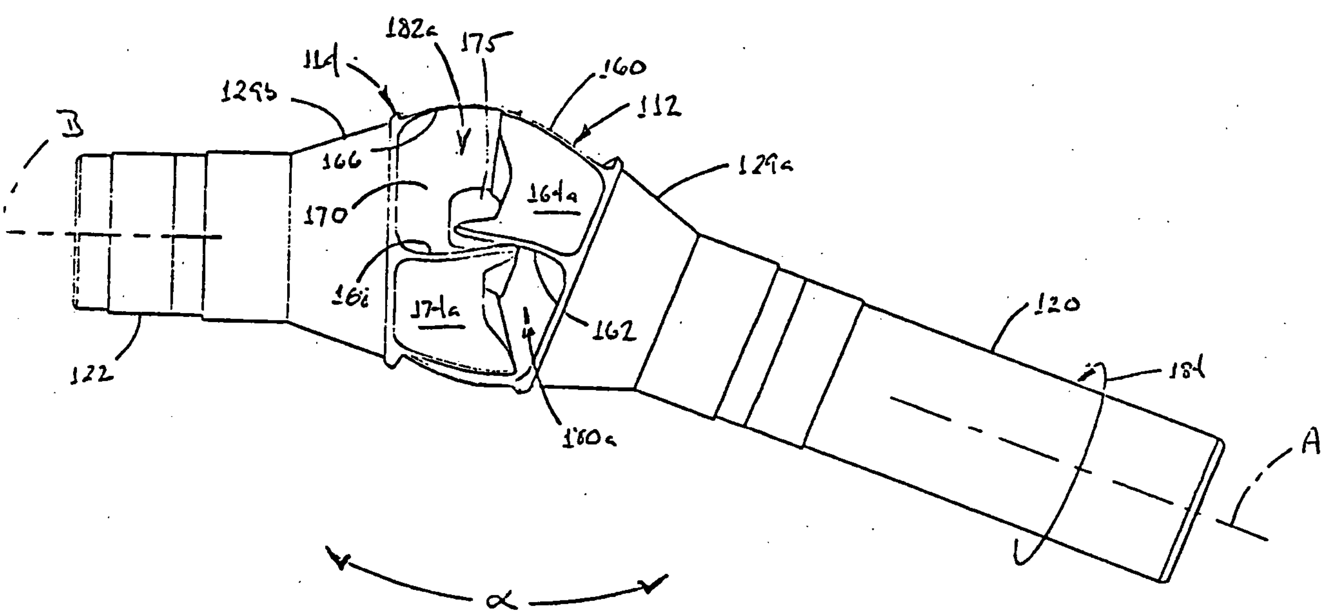

B
122
129b
114
182a
175
160
112
166
170
164a
129a
168
174a
162
180a
120
184
A
α

FIG. 7B

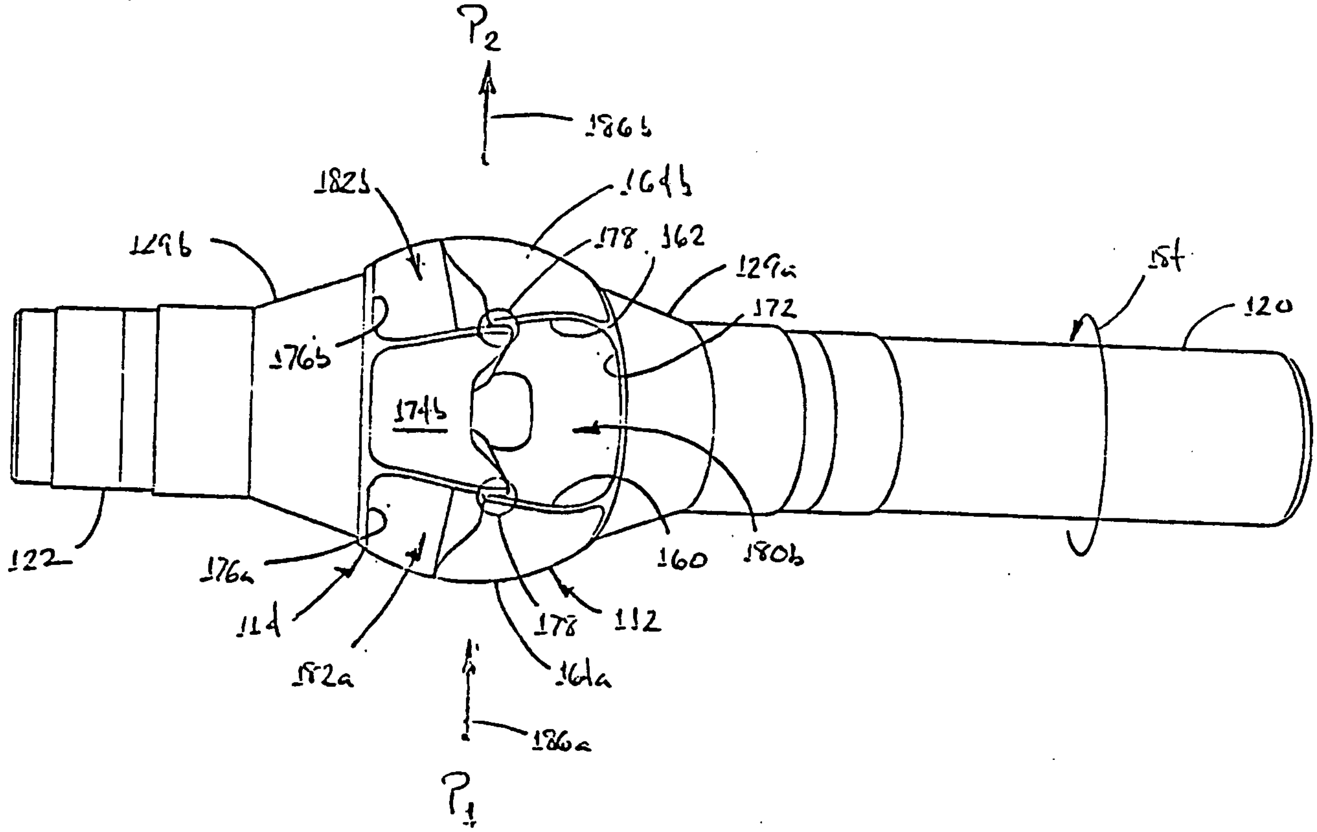

P2
186b
182b
164b
178
162
129b
129a
154
120
172
176b
174b
122
176a
114
182a
180b
160
178
112
164a
186a
P1

FIG. 7C

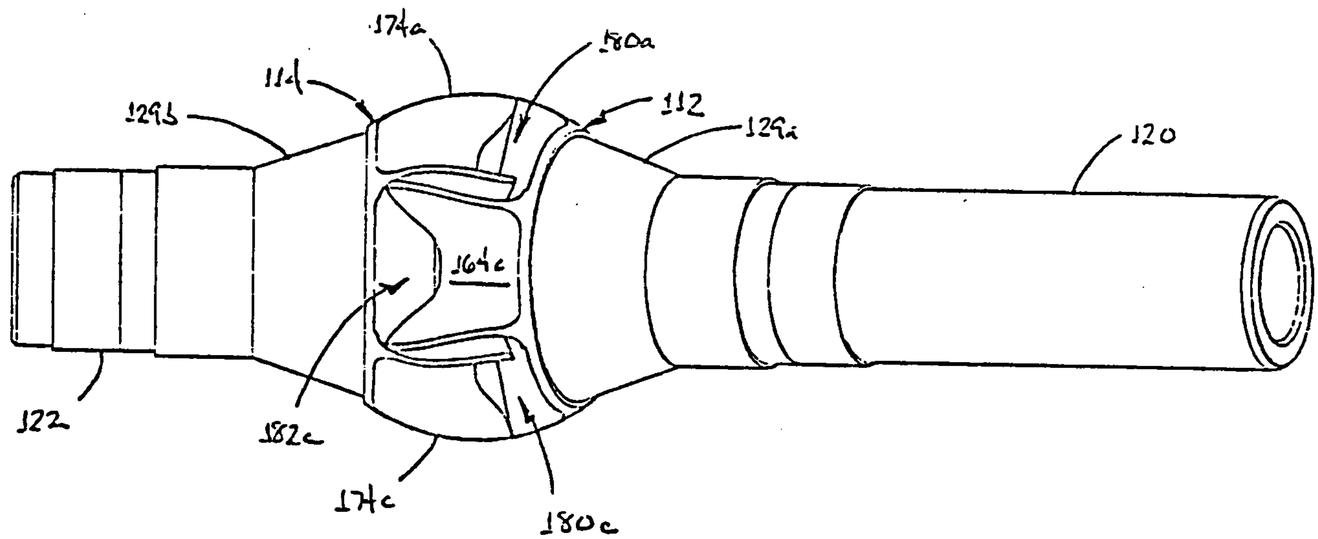


FIG. 8A

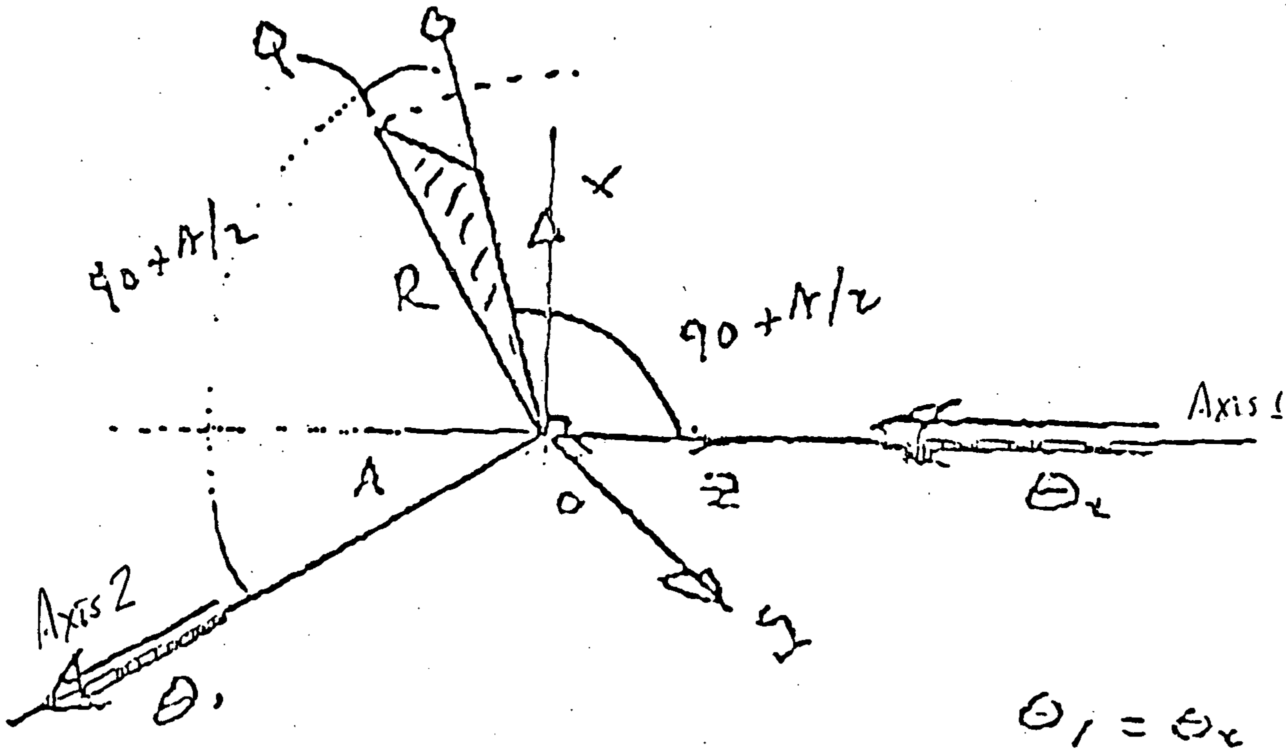


FIG 8B

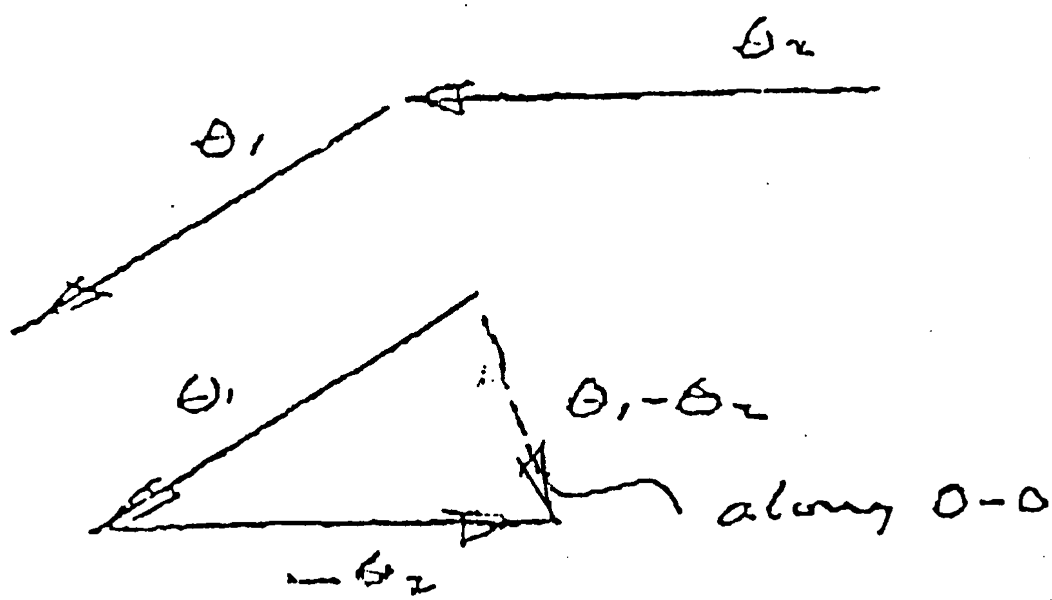


FIG 8C

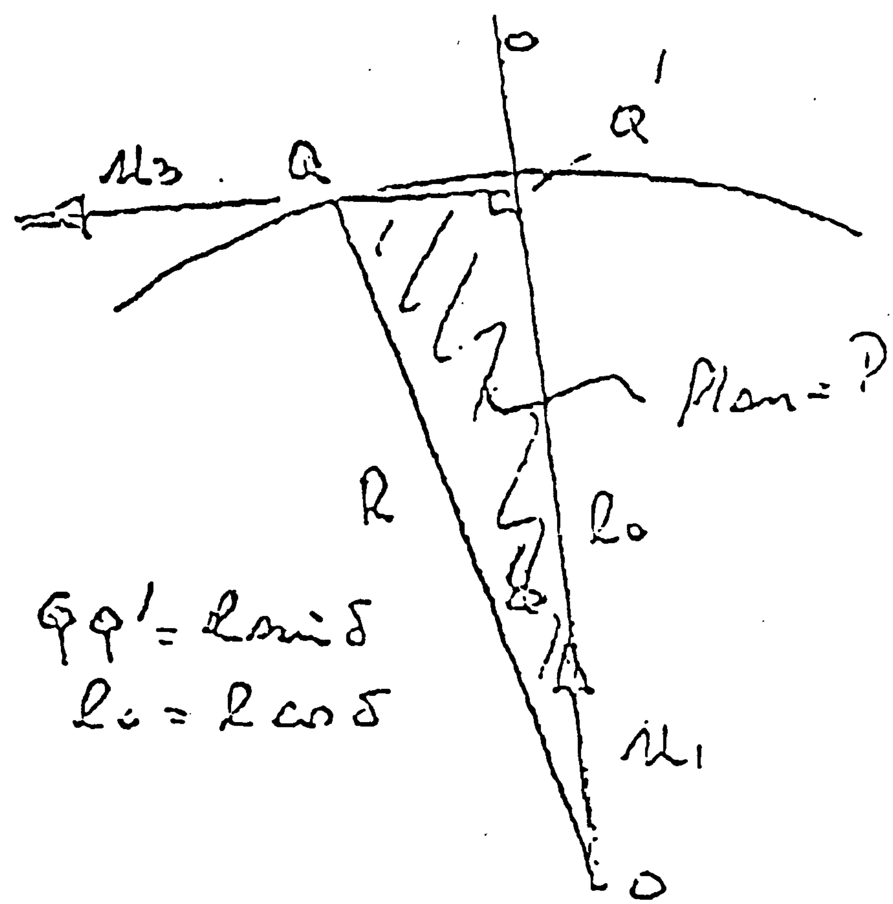


FIG. 8D

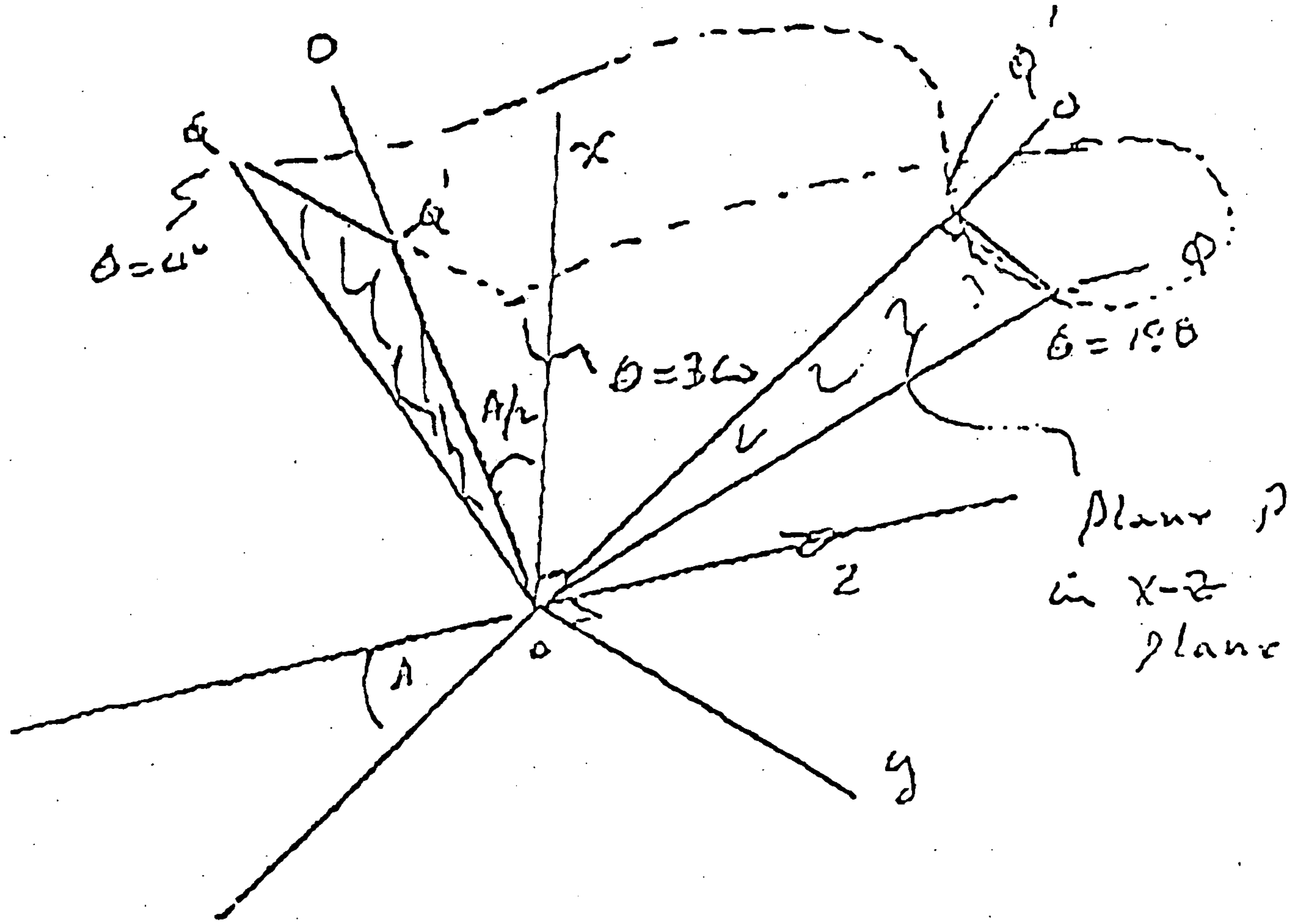


FIG. 8E

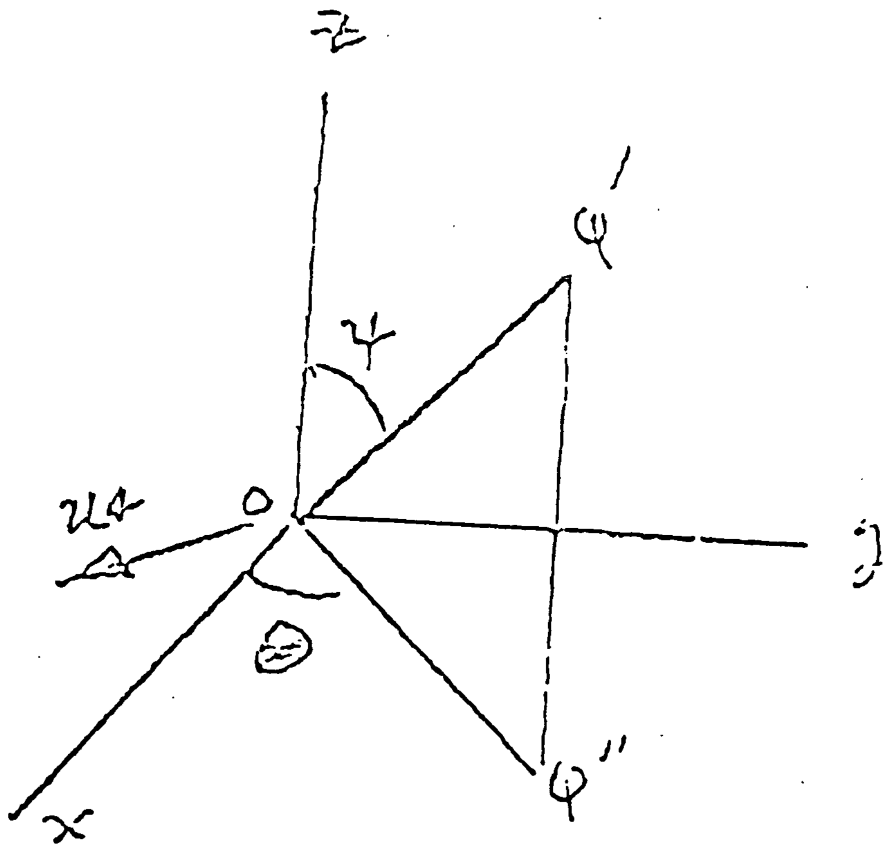


FIG. 9A

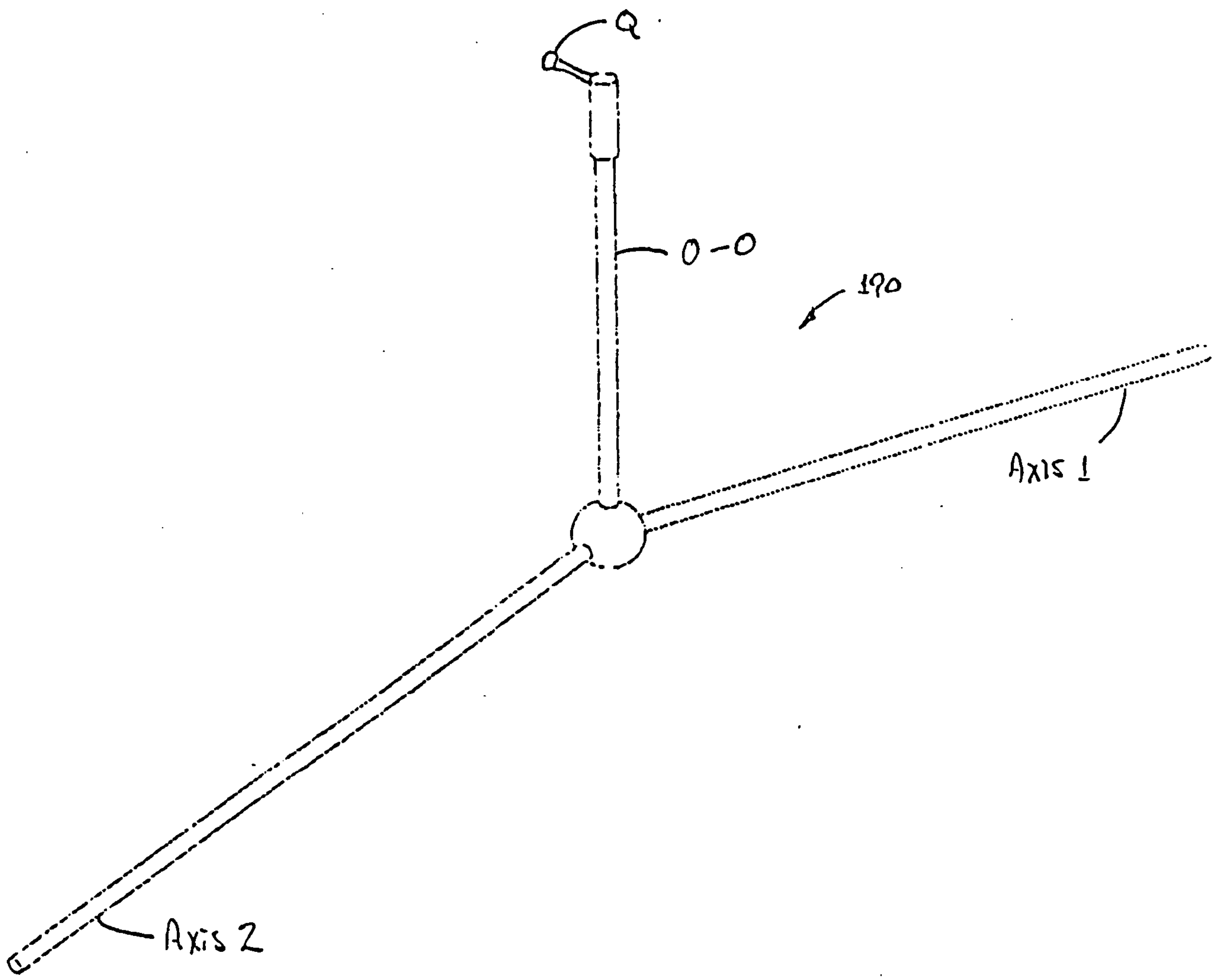


FIG. 9B

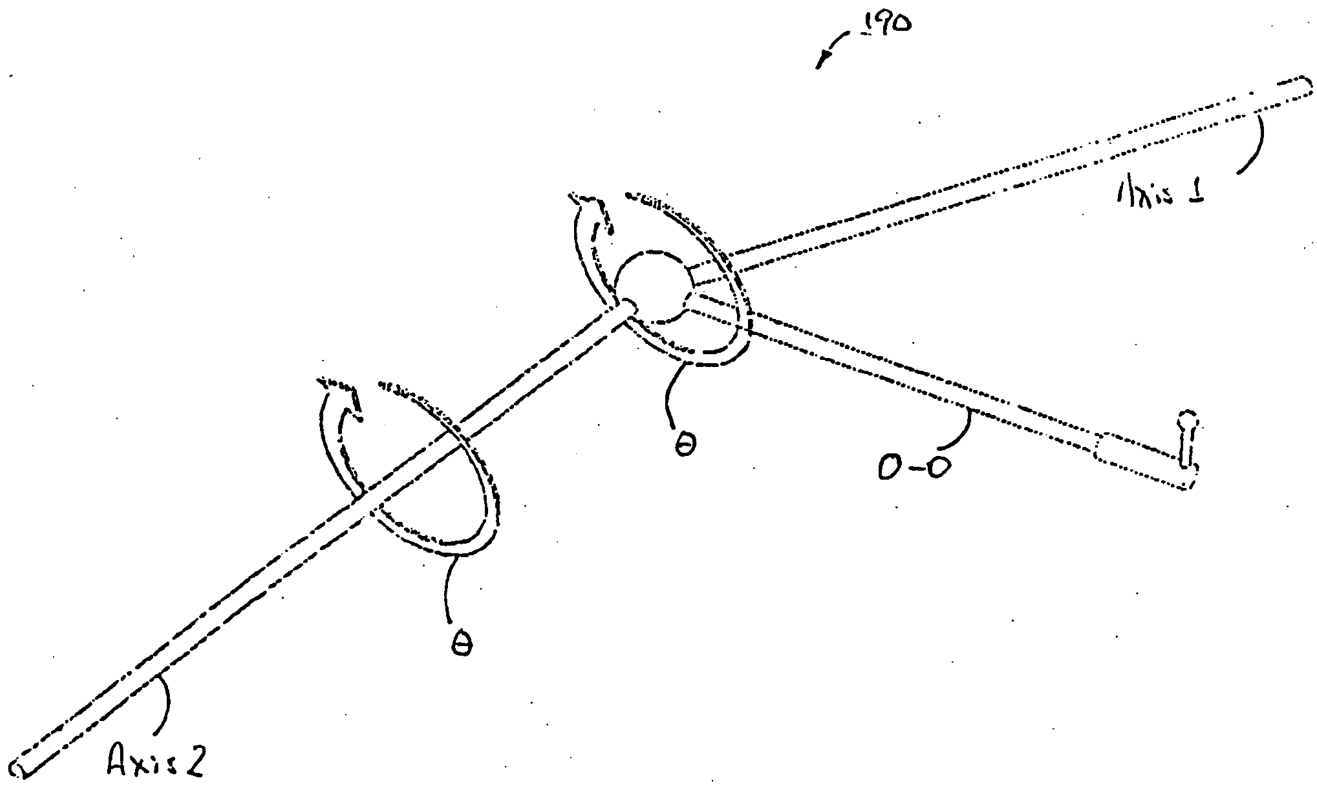


FIG. 9C

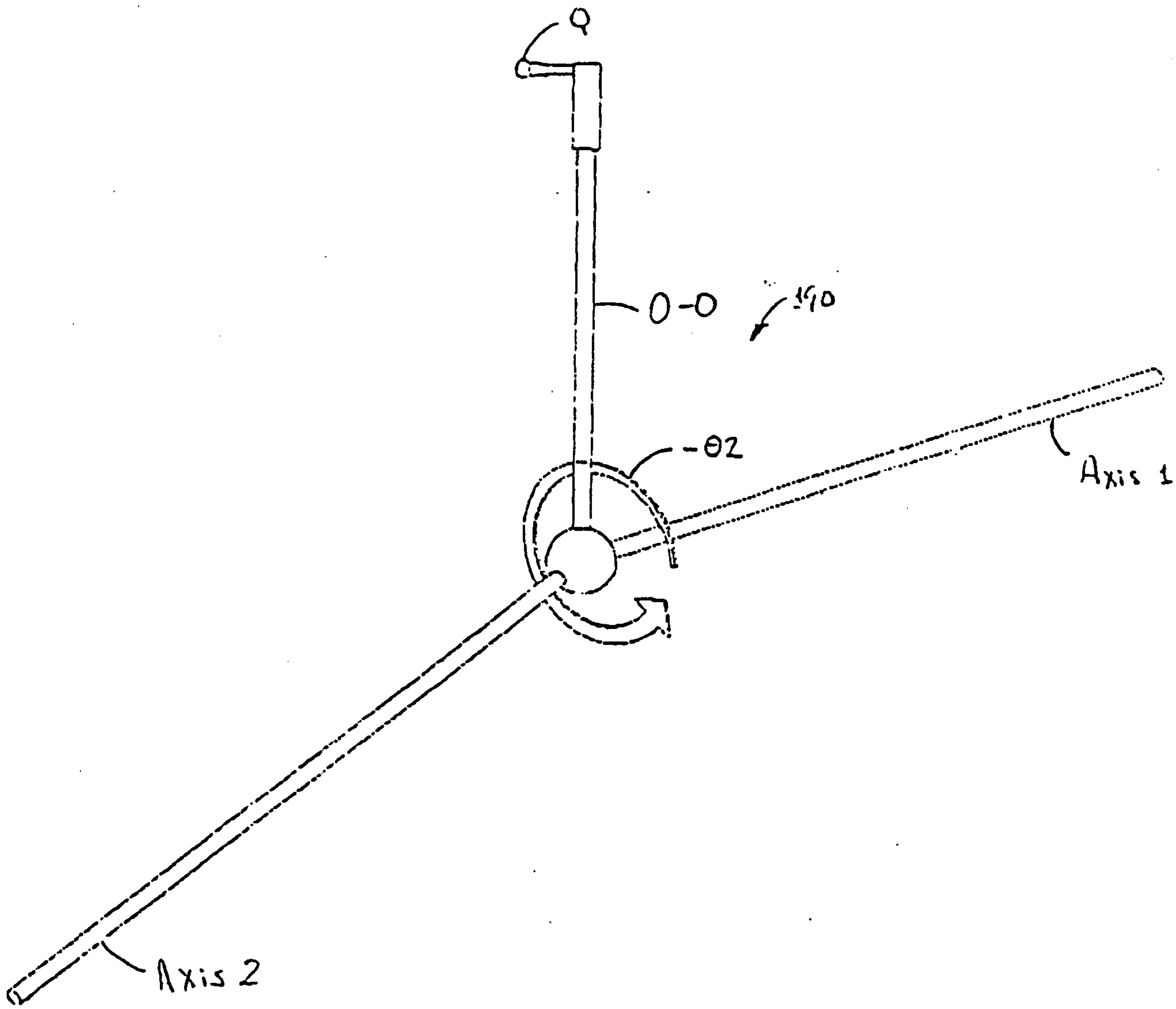


FIG. 9D

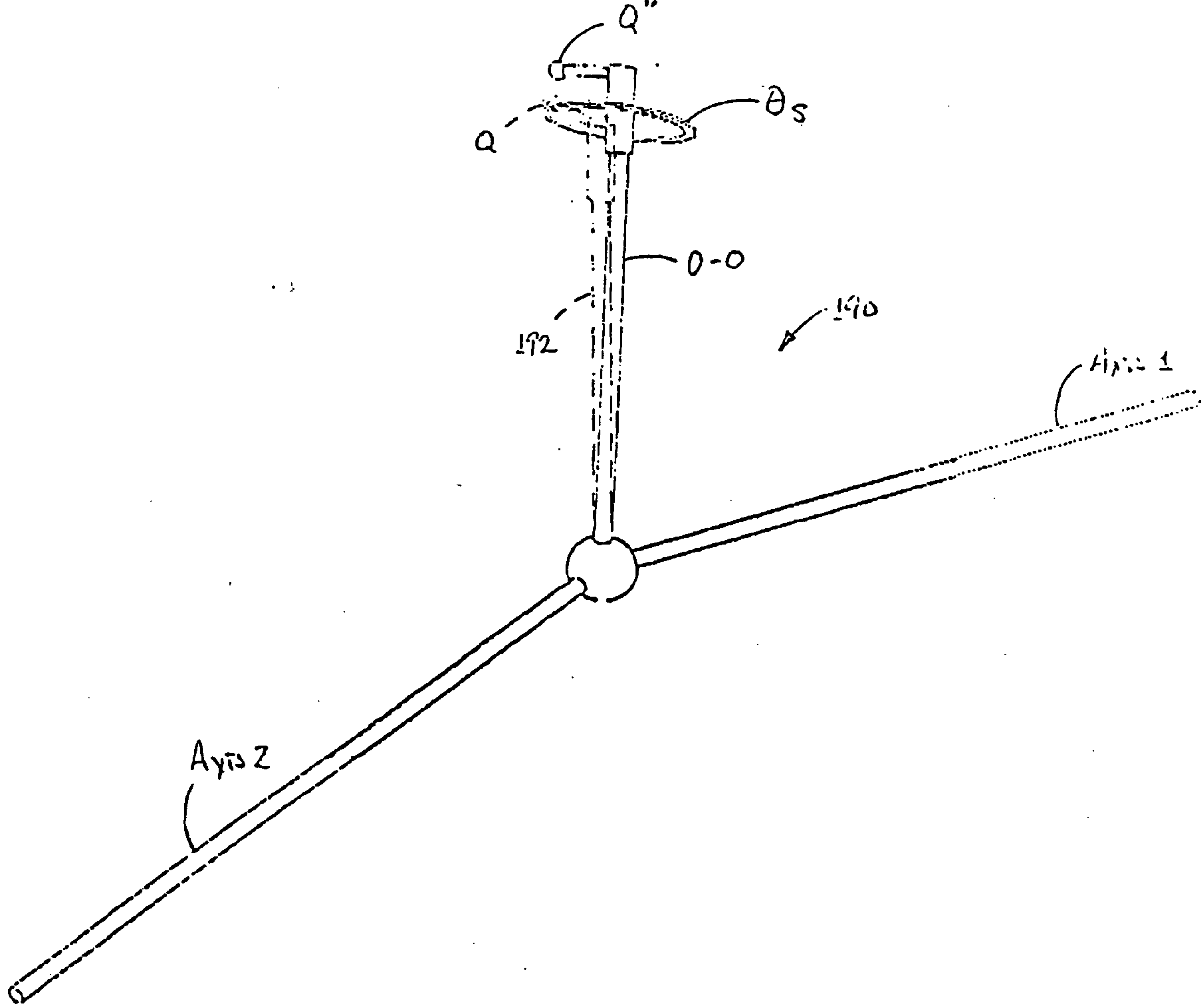


FIG. 10A

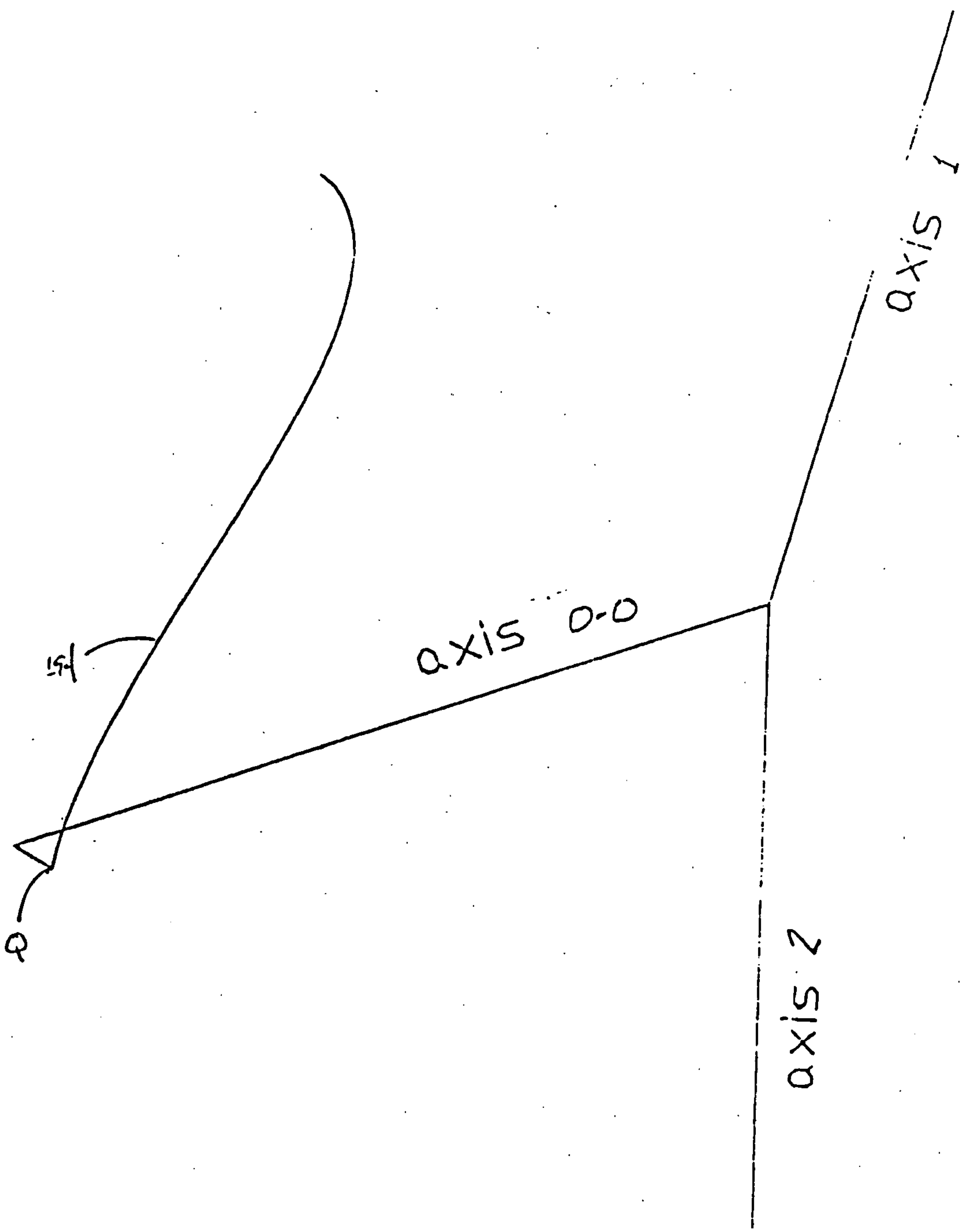


FIG. 10B

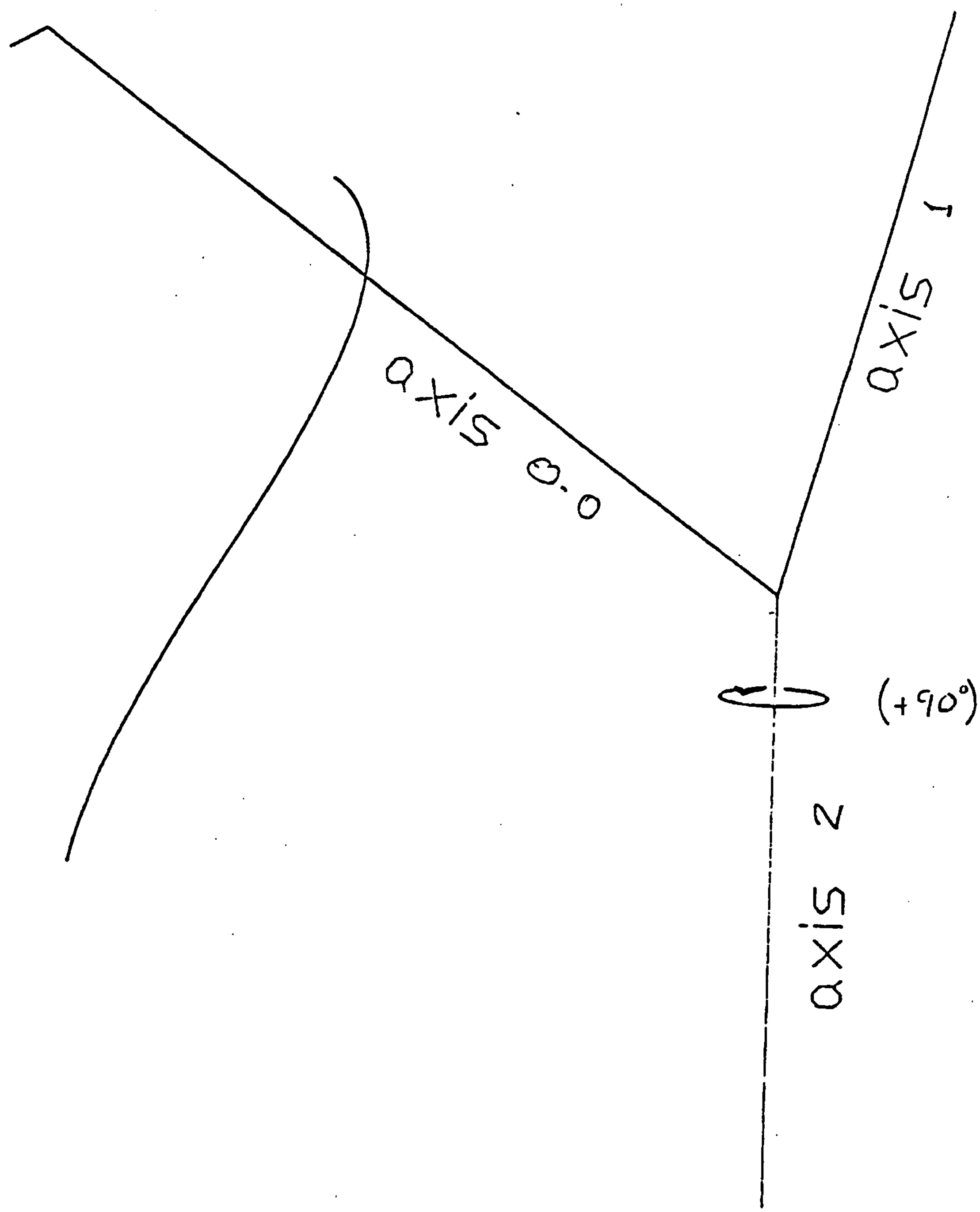
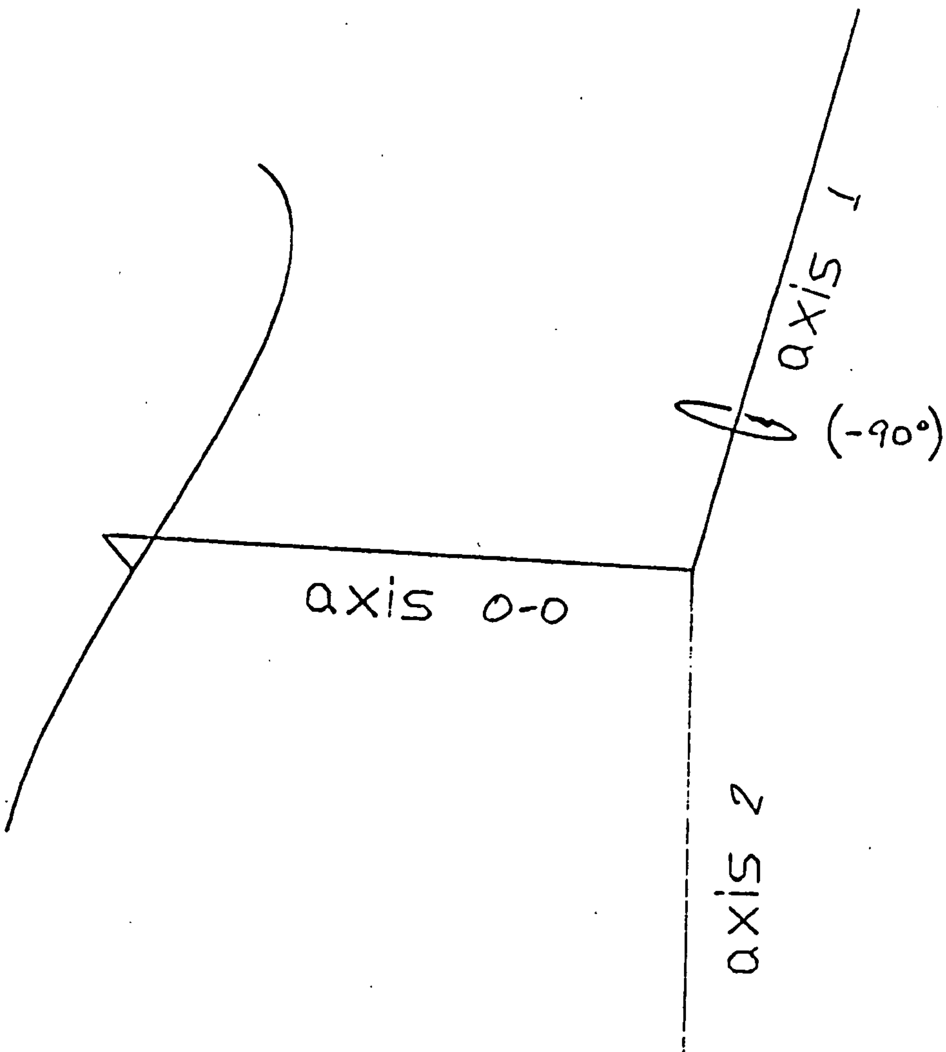
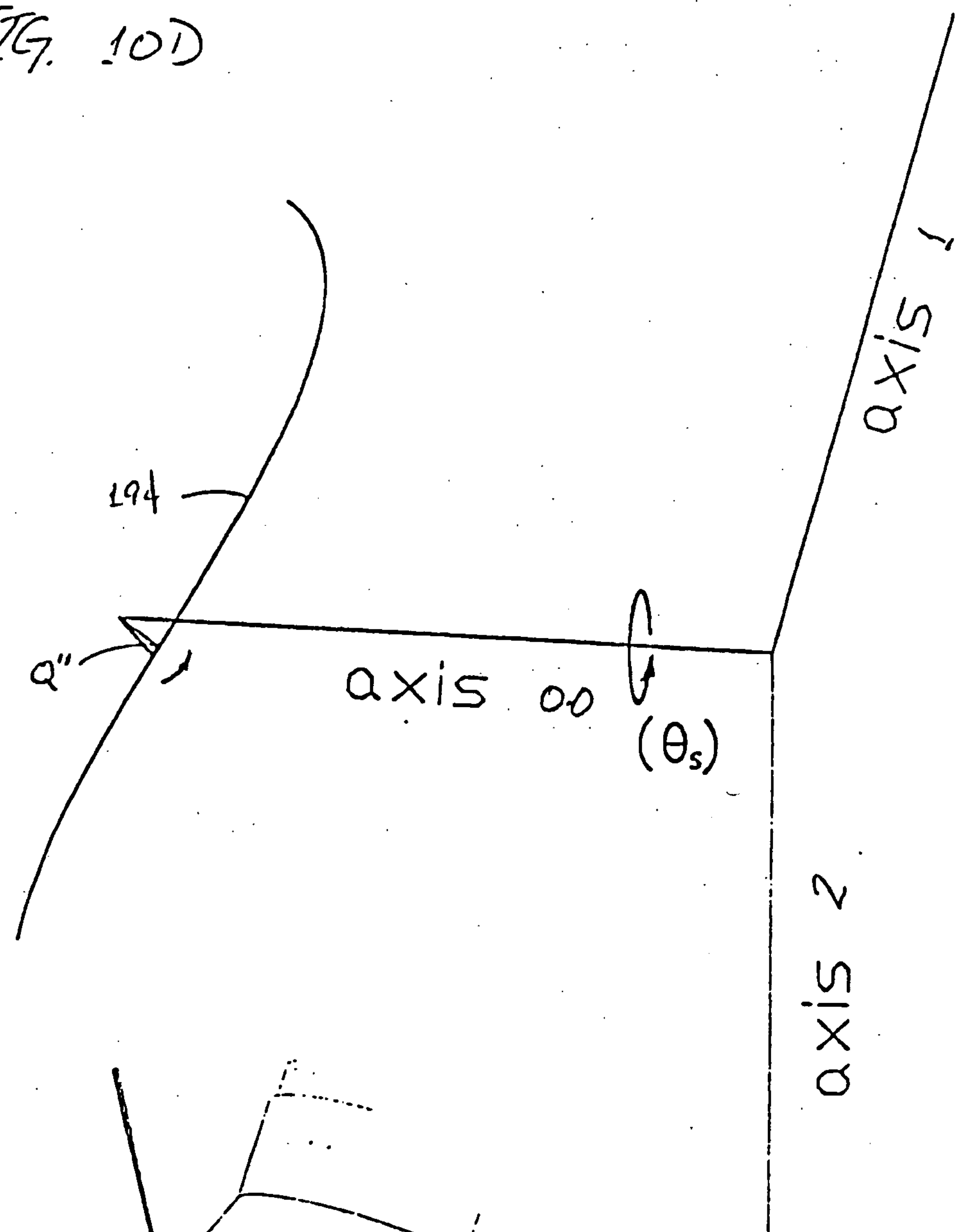


FIG. 10C



STEP 2

FIG. 10D



STEP 3

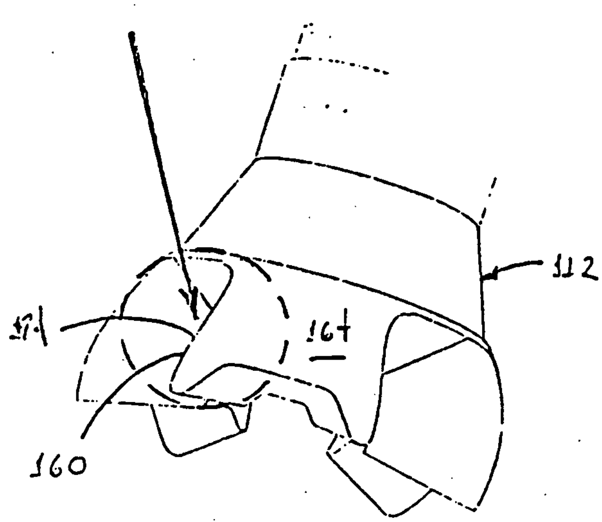


FIG. 10E

FIG. 11A

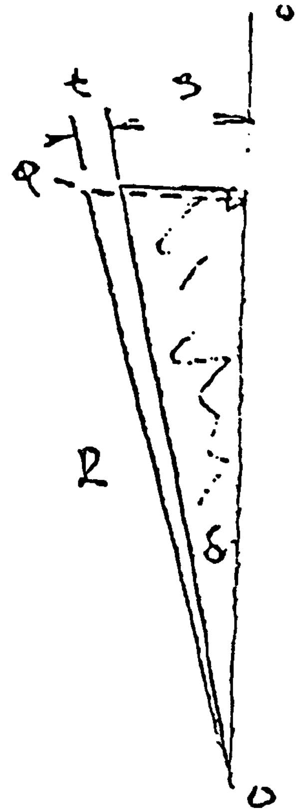


FIG 11B

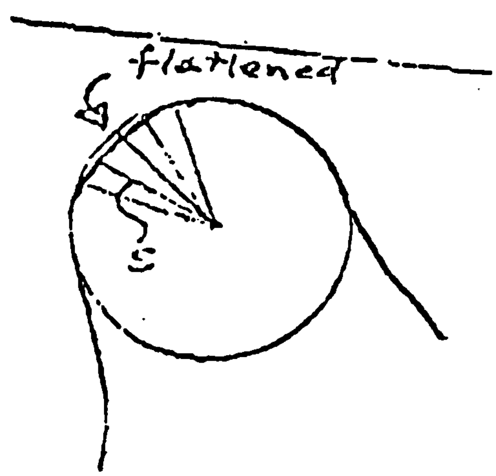


FIG. 12

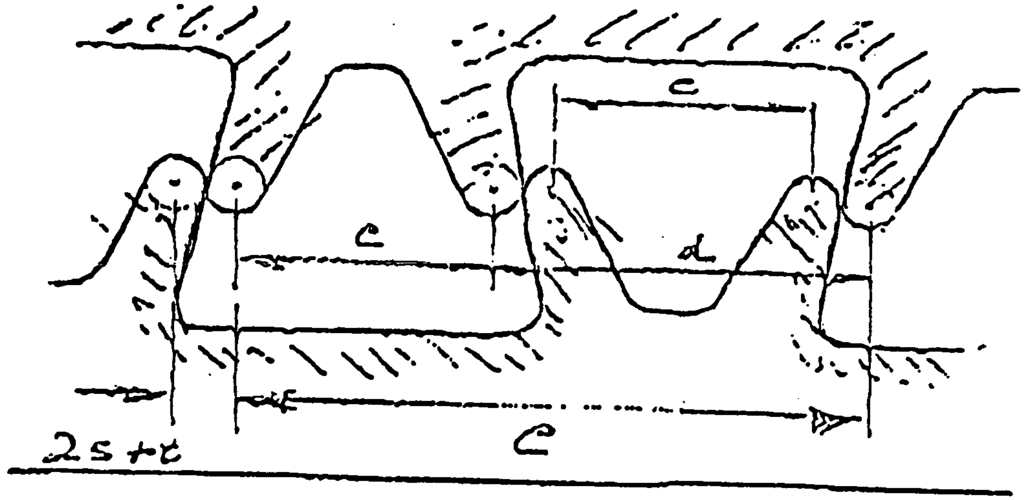


FIG. 13A

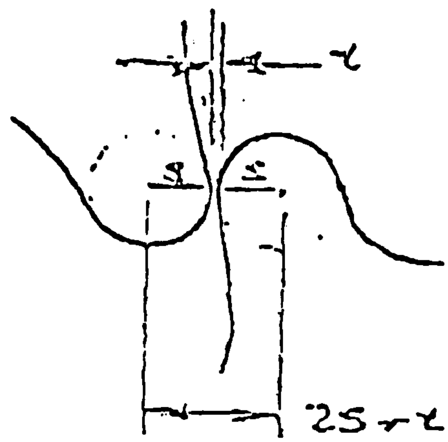
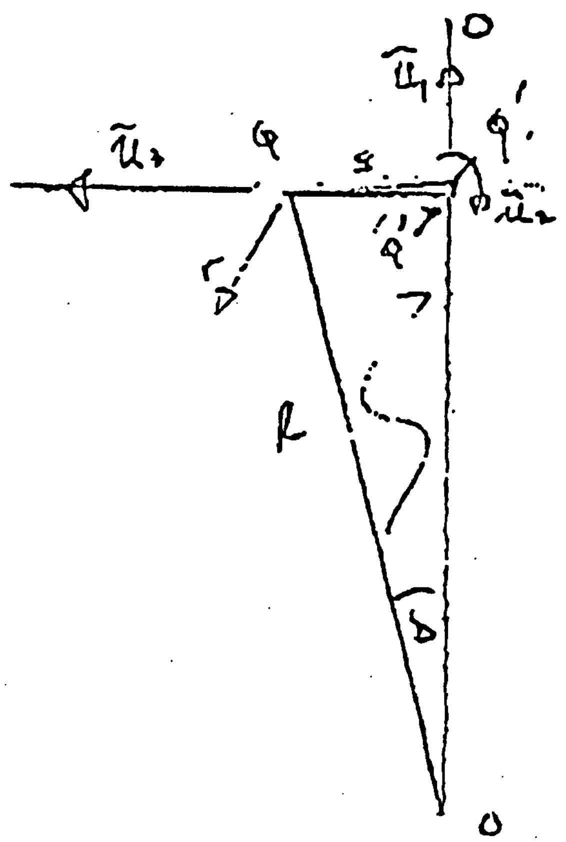


FIG. 13B



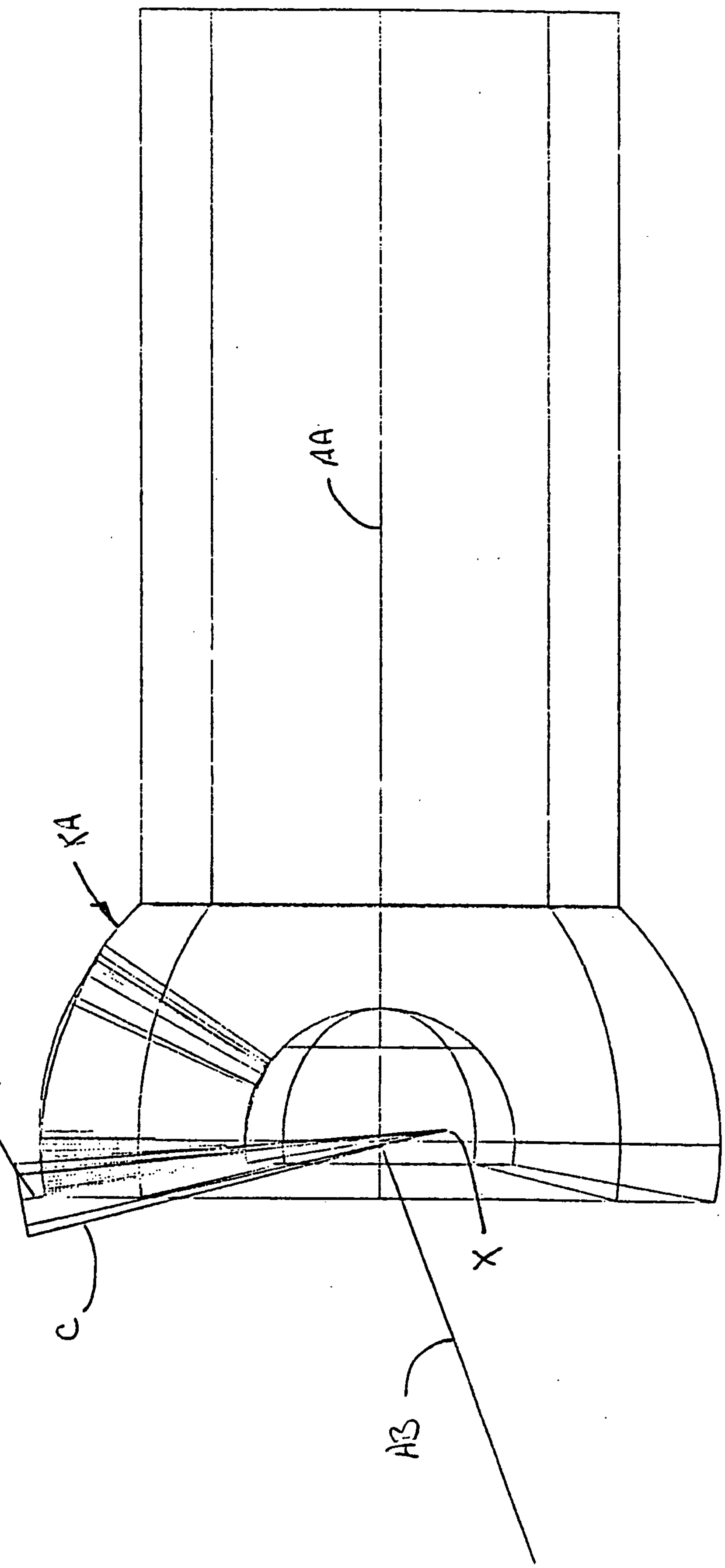


FIG. 14

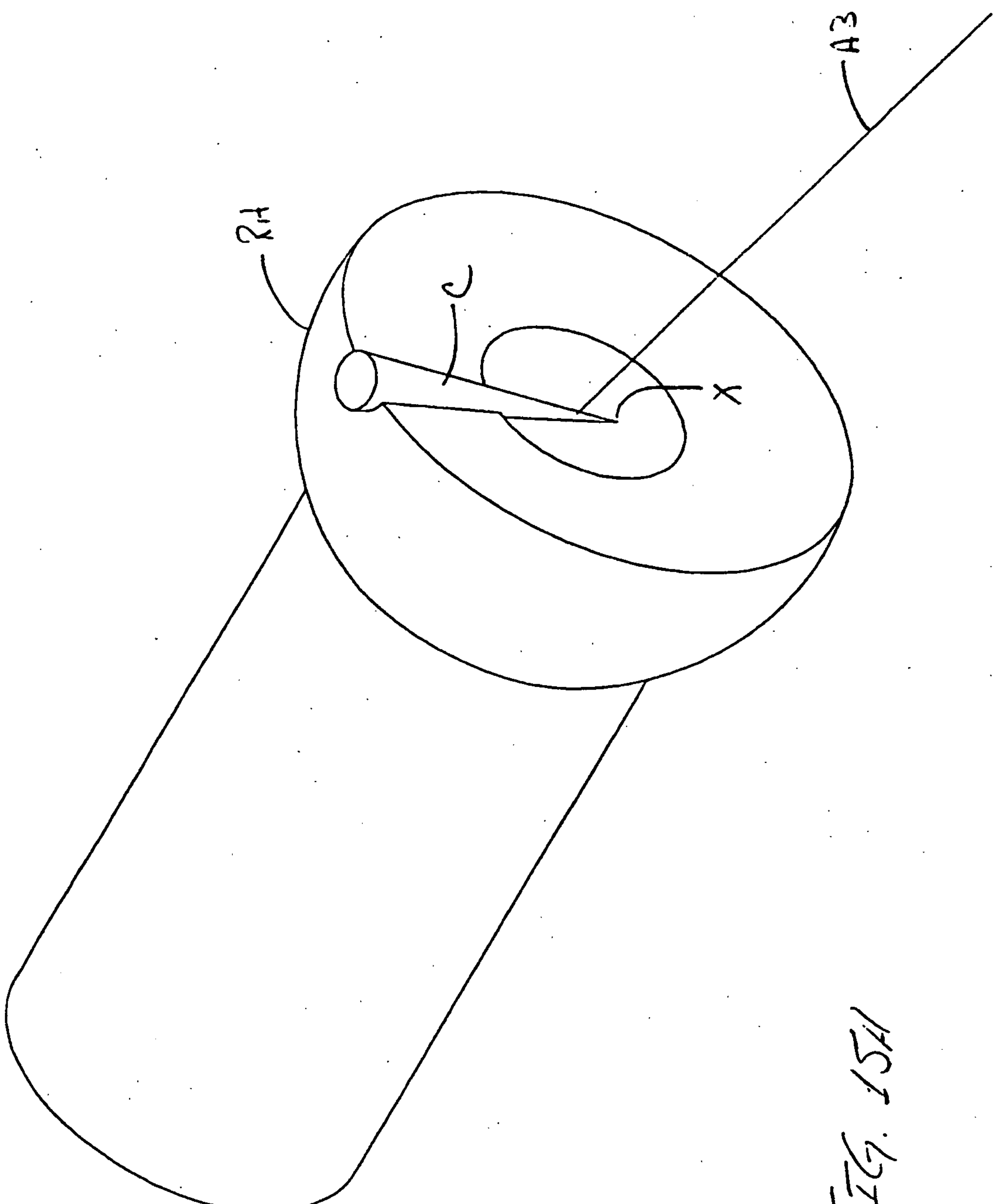


FIG. 15A

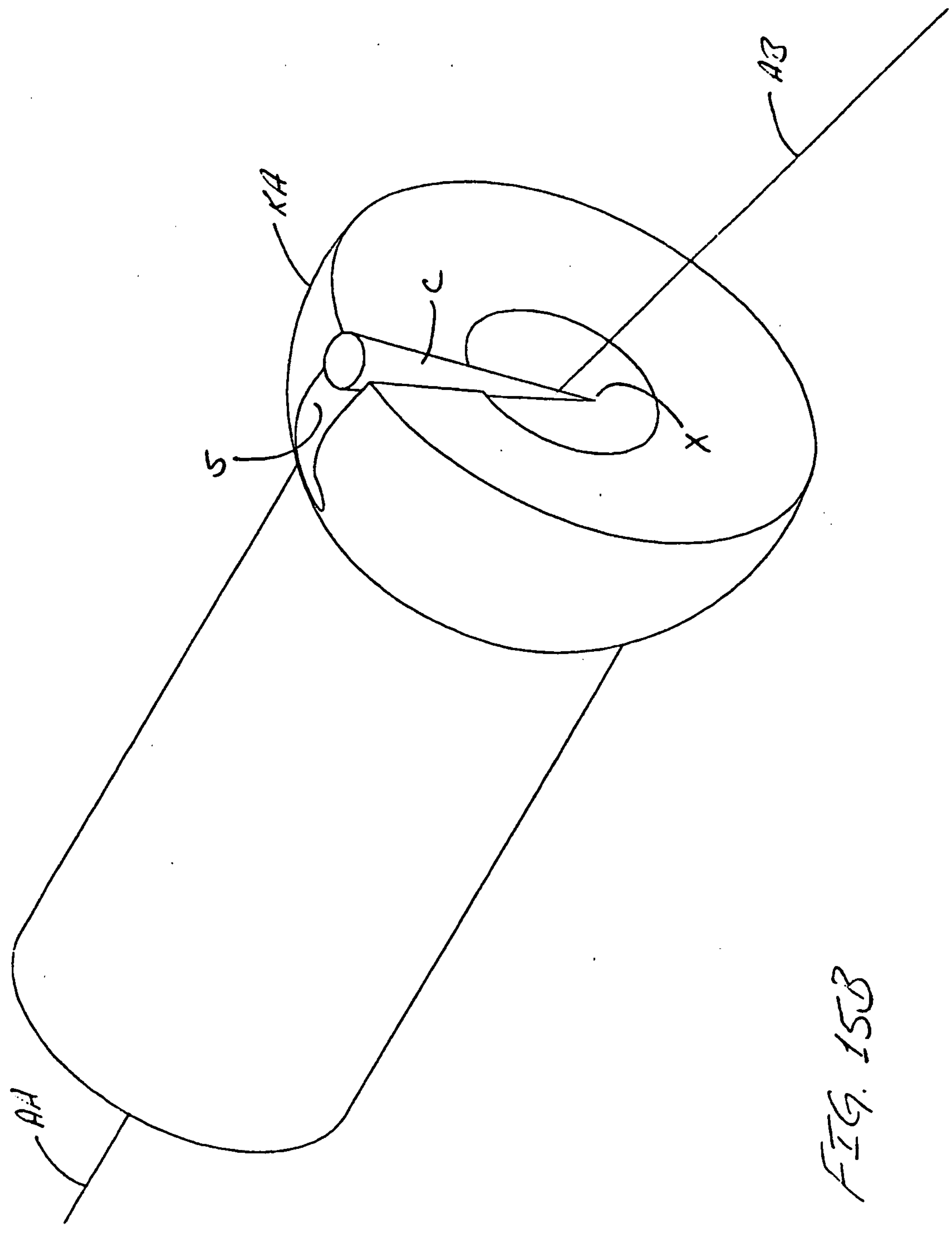


FIG. 15B

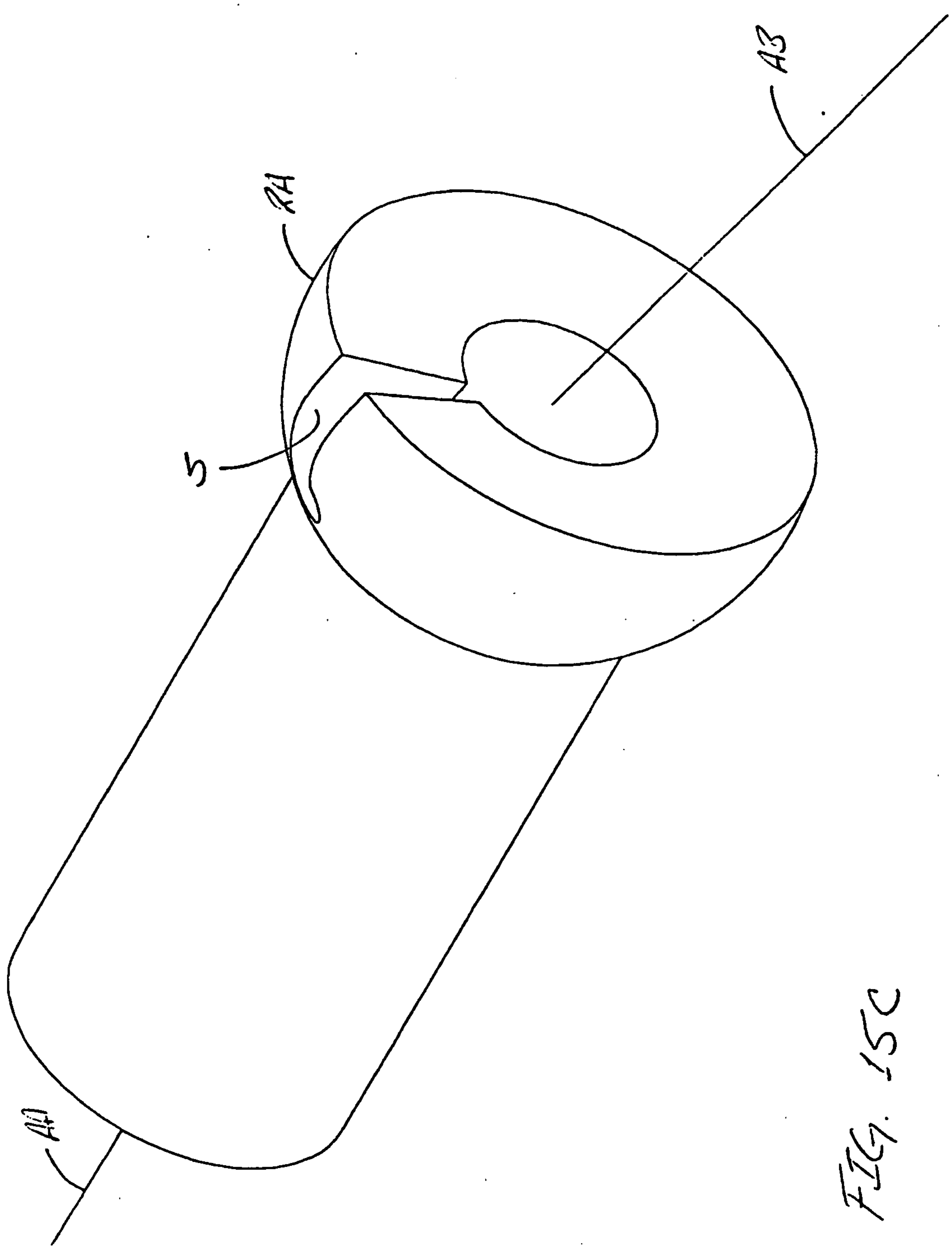


FIG. 15C

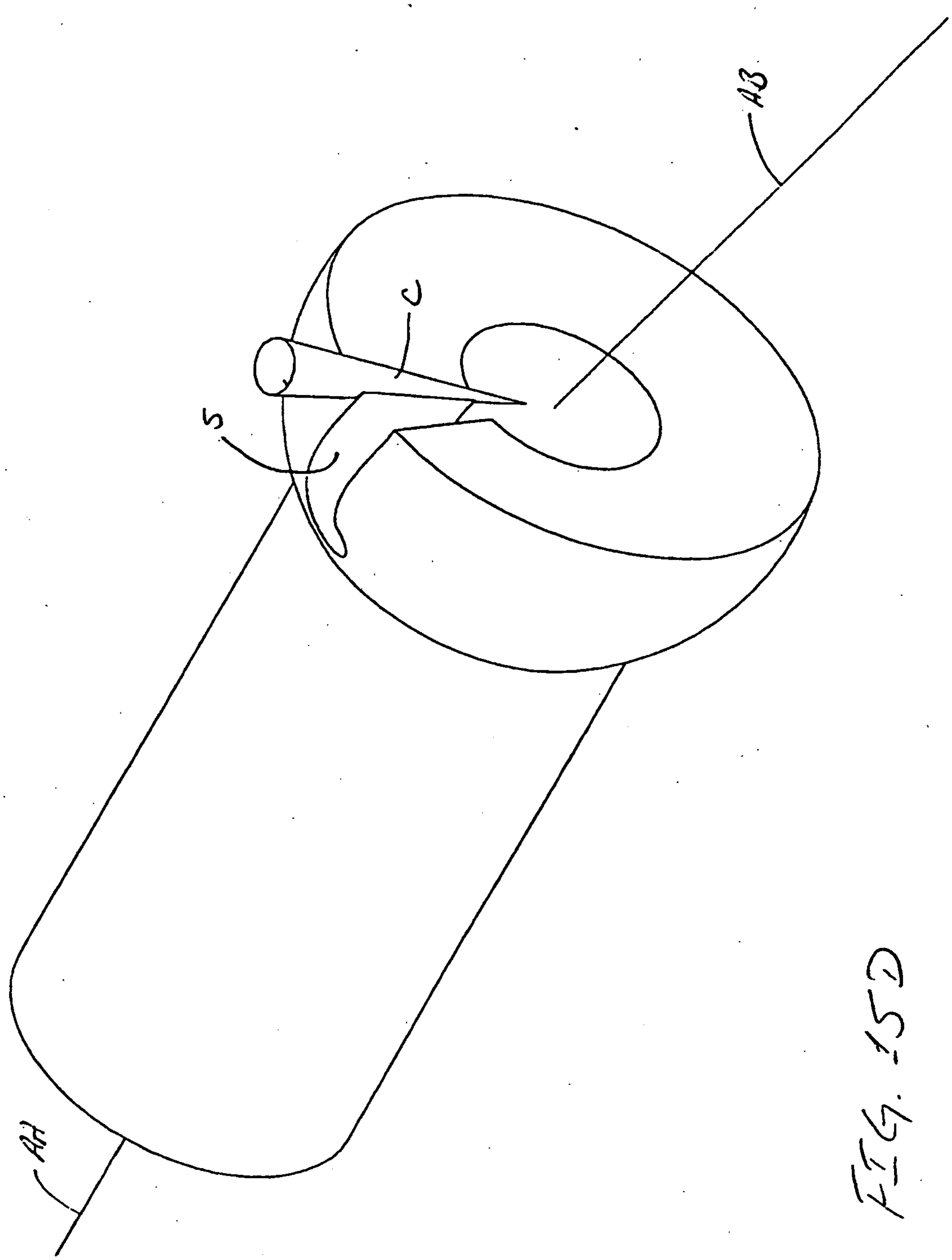


FIG. 15D

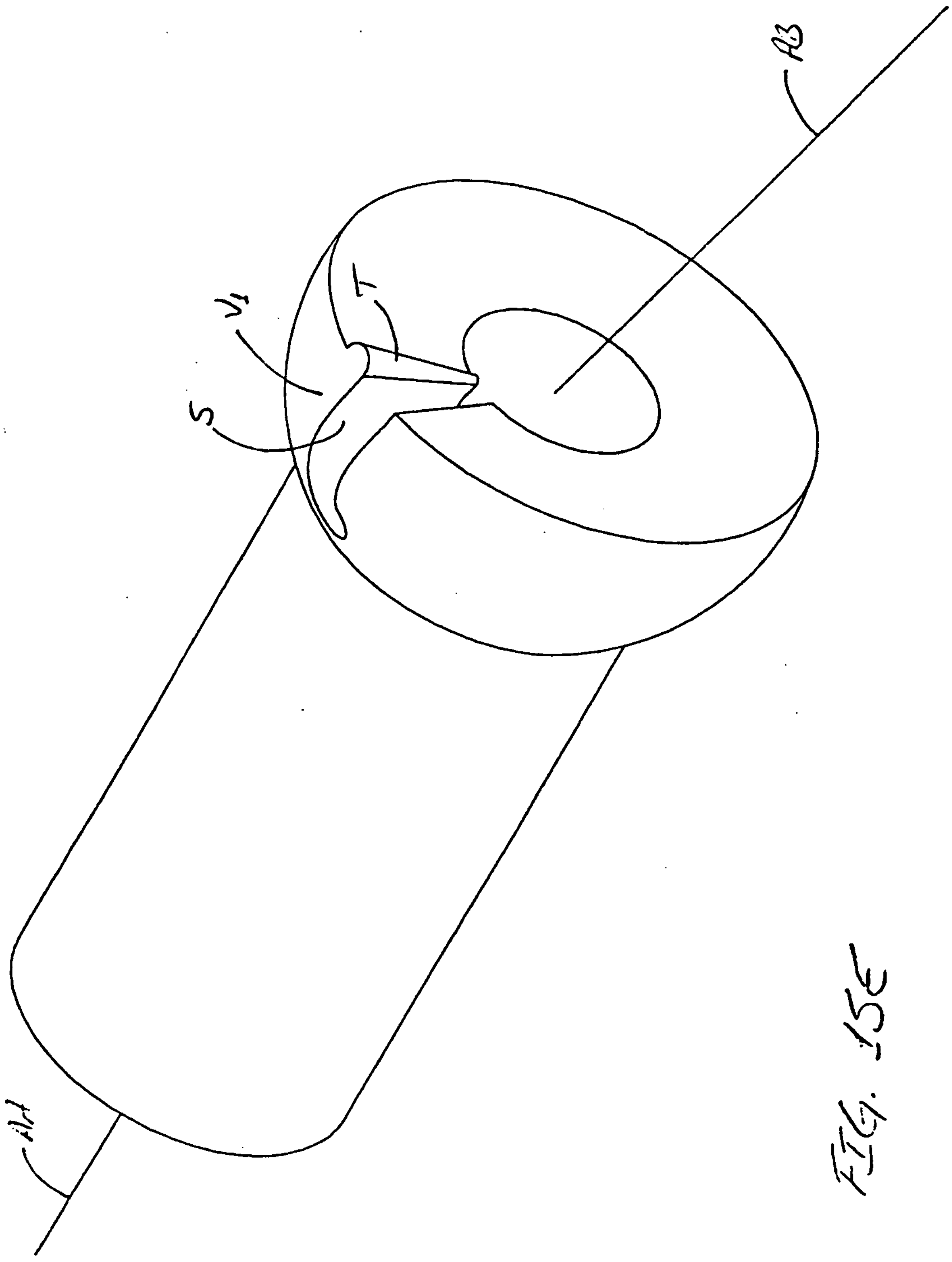


FIG. 15E

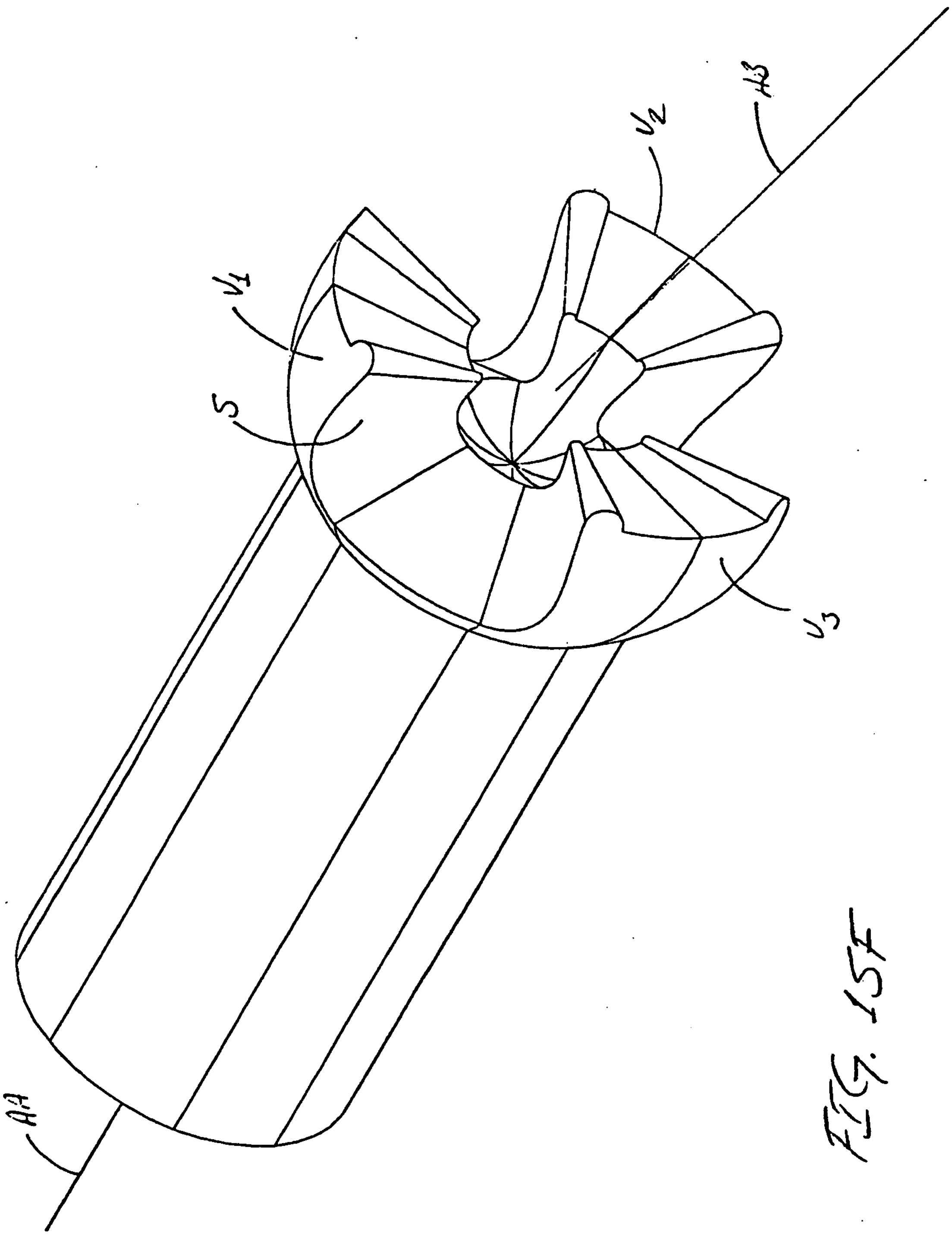


FIG. 15F

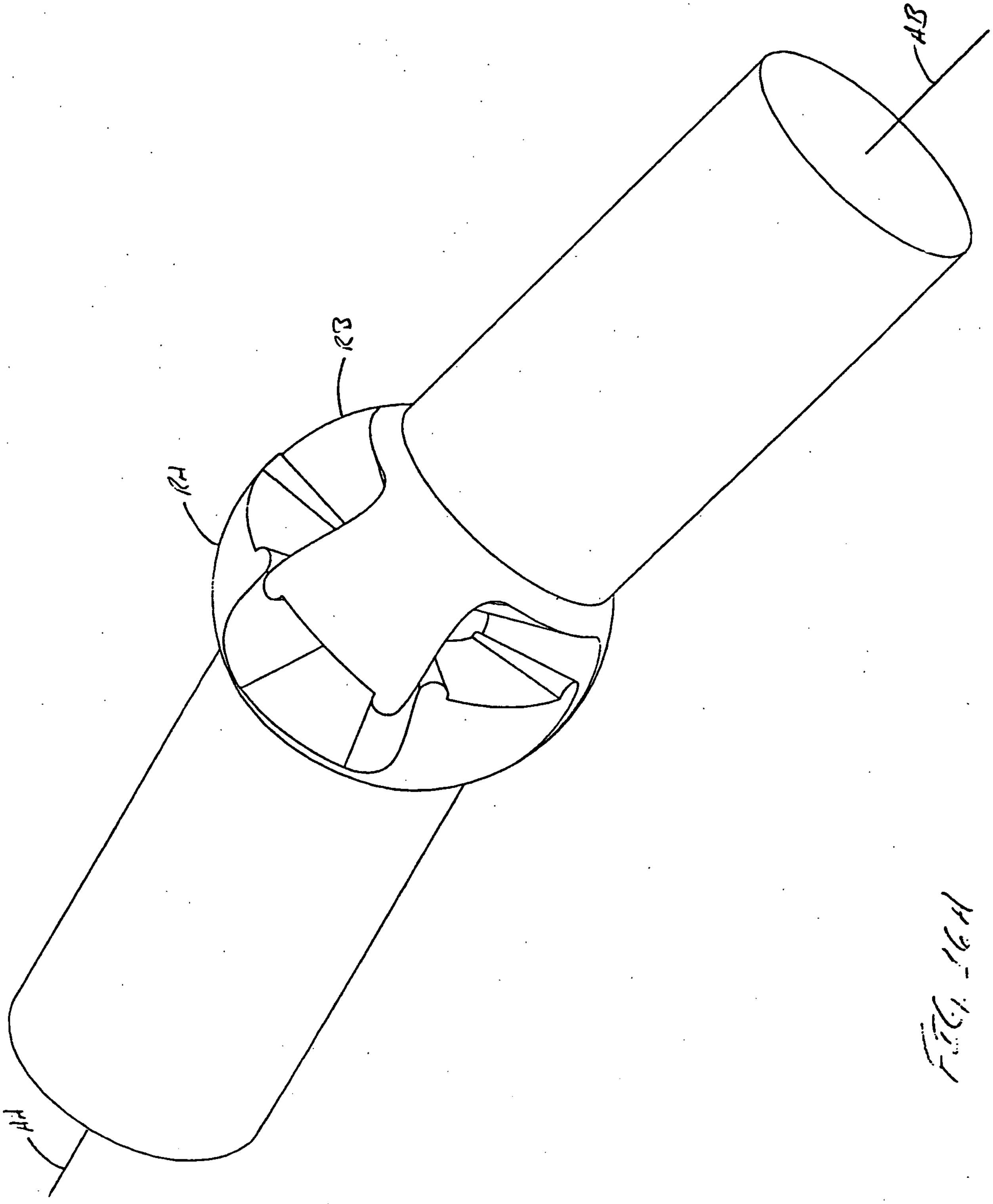


FIG. 16A

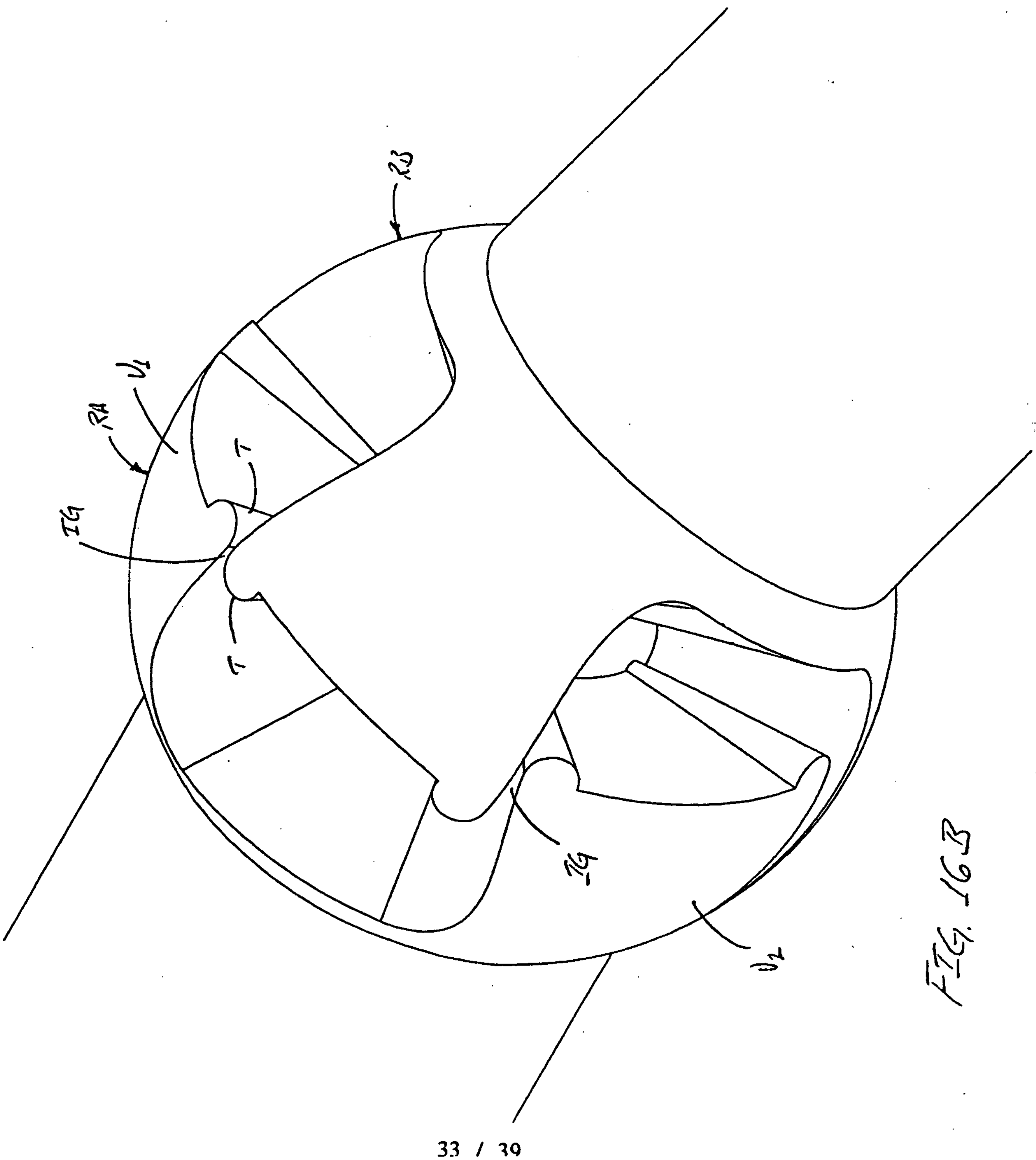
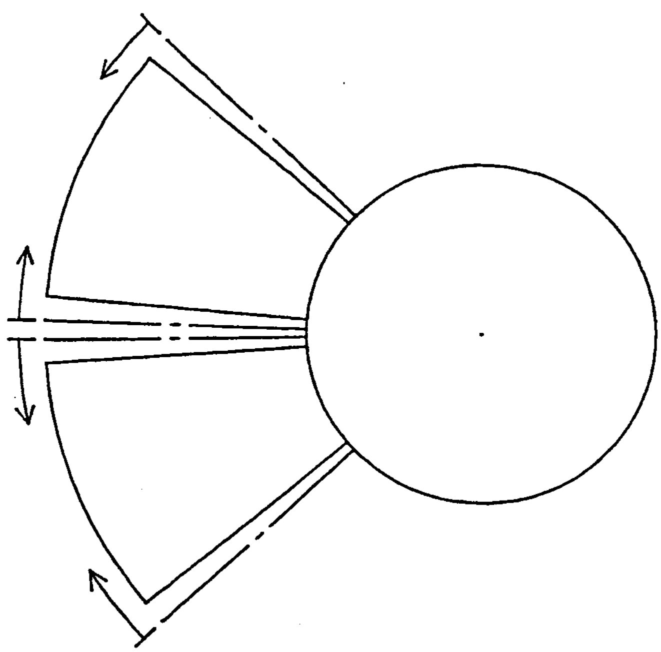


FIG. 16B



SEAL SURFACE ROTATION
RELATIVE TO OTHER SURFACES

FIG. 17

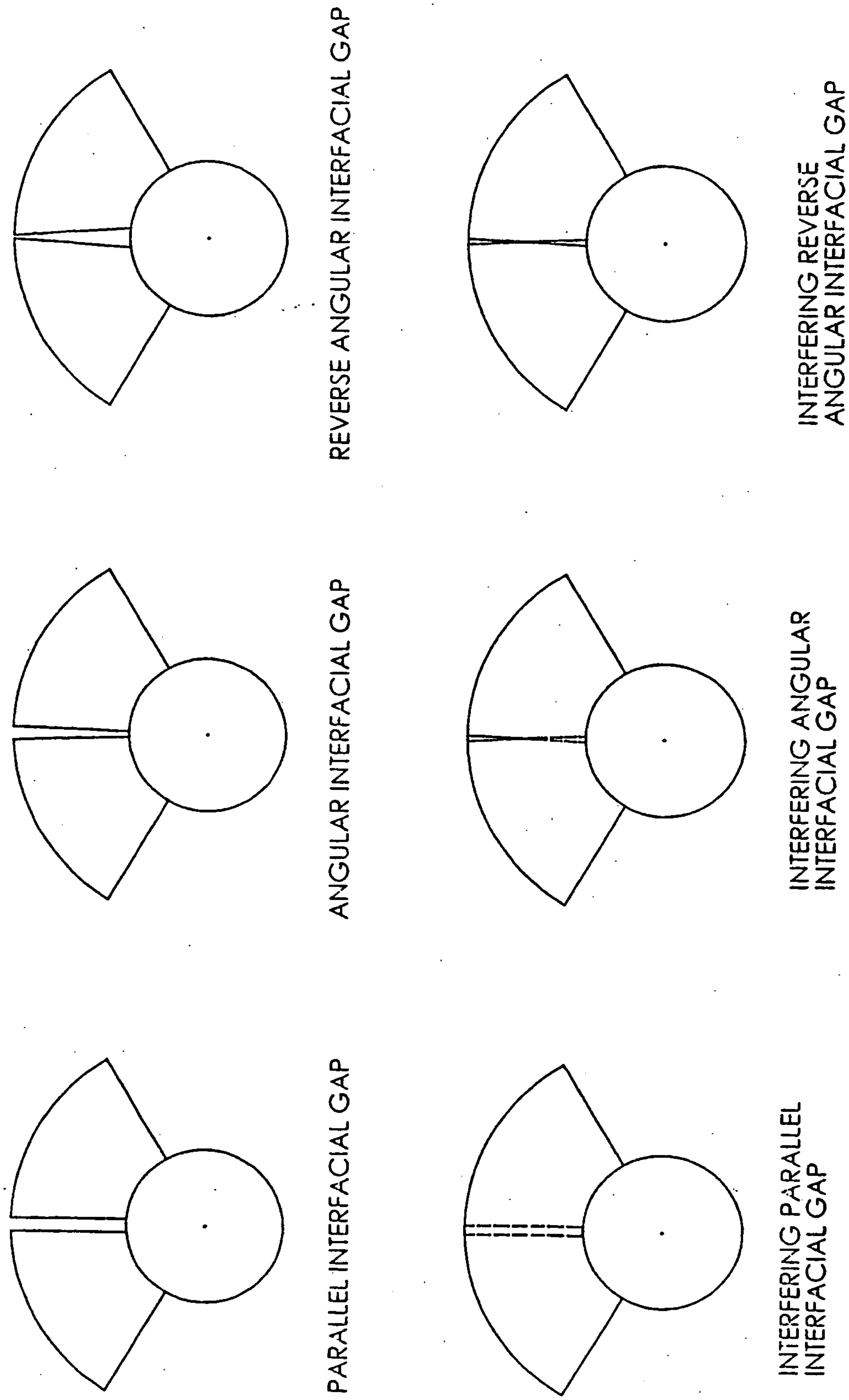

PARALLEL INTERFACIAL GAP
ANGULAR INTERFACIAL GAP
REVERSE ANGULAR INTERFACIAL GAP
INTERFERING PARALLEL INTERFACIAL GAP
INTERFERING ANGULAR INTERFACIAL GAP
INTERFERING REVERSE ANGULAR INTERFACIAL GAP

FIG. 18

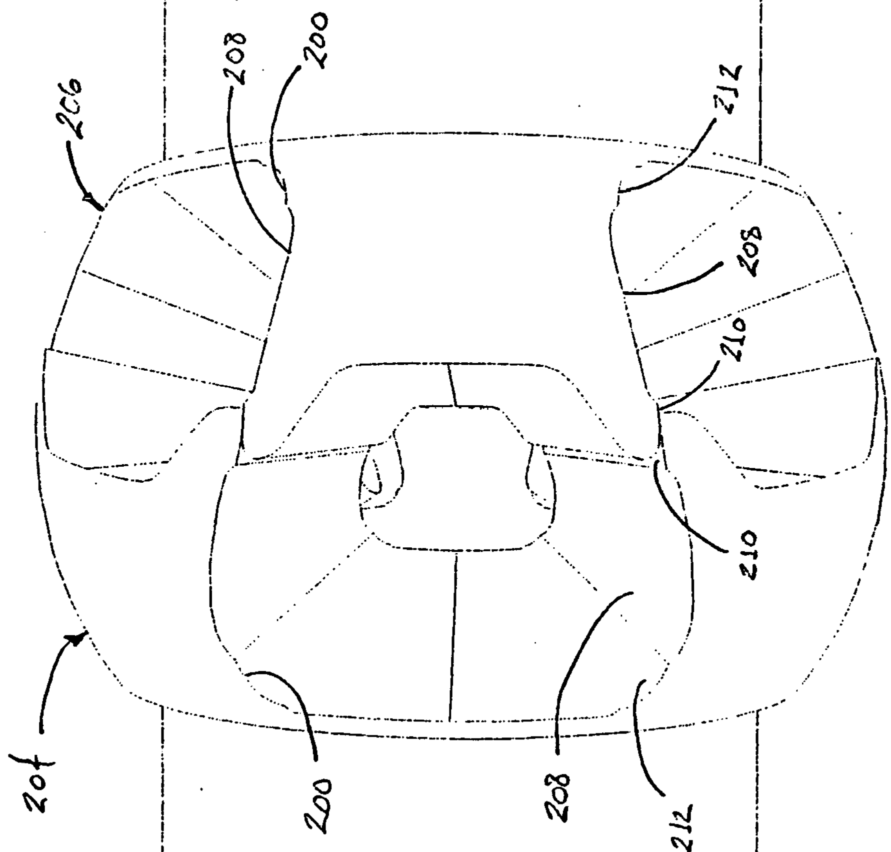


FIG. 19A

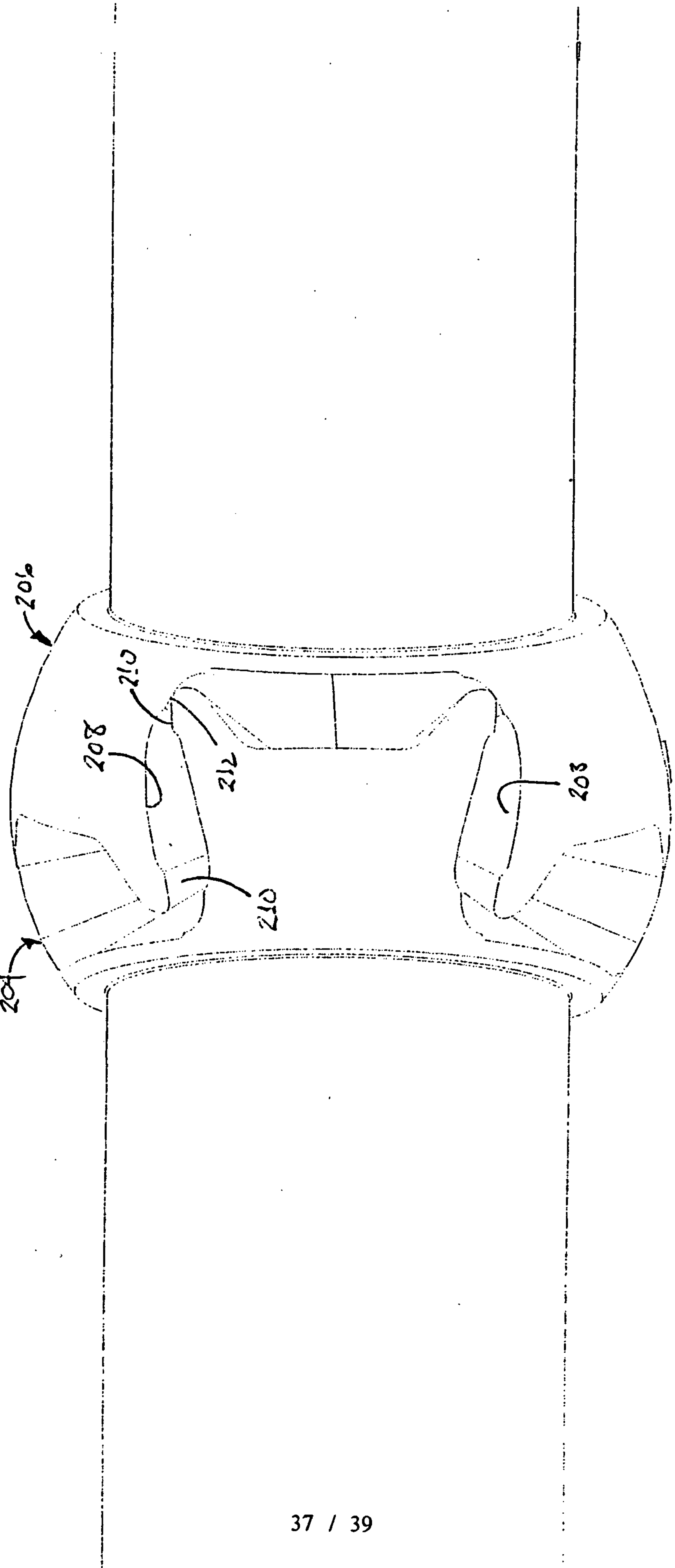


FIG. 19B

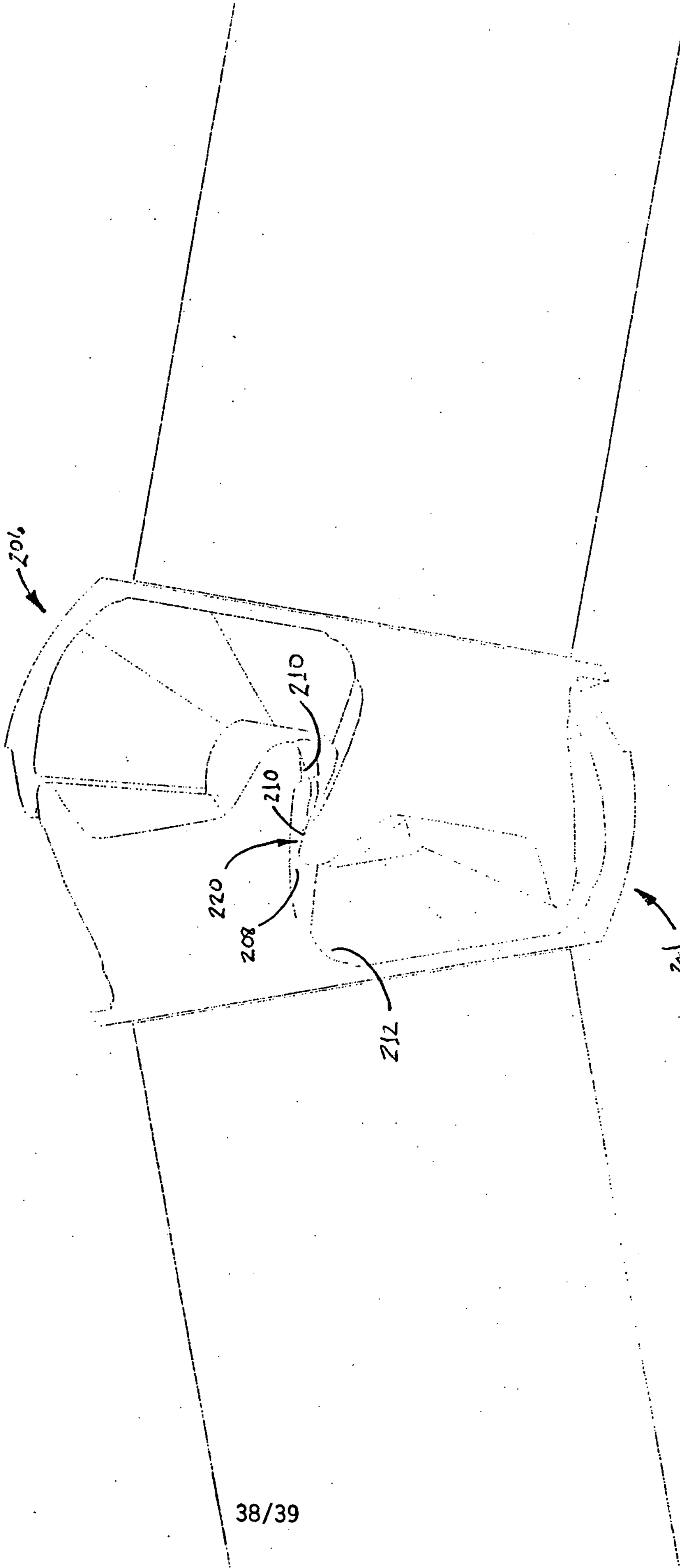
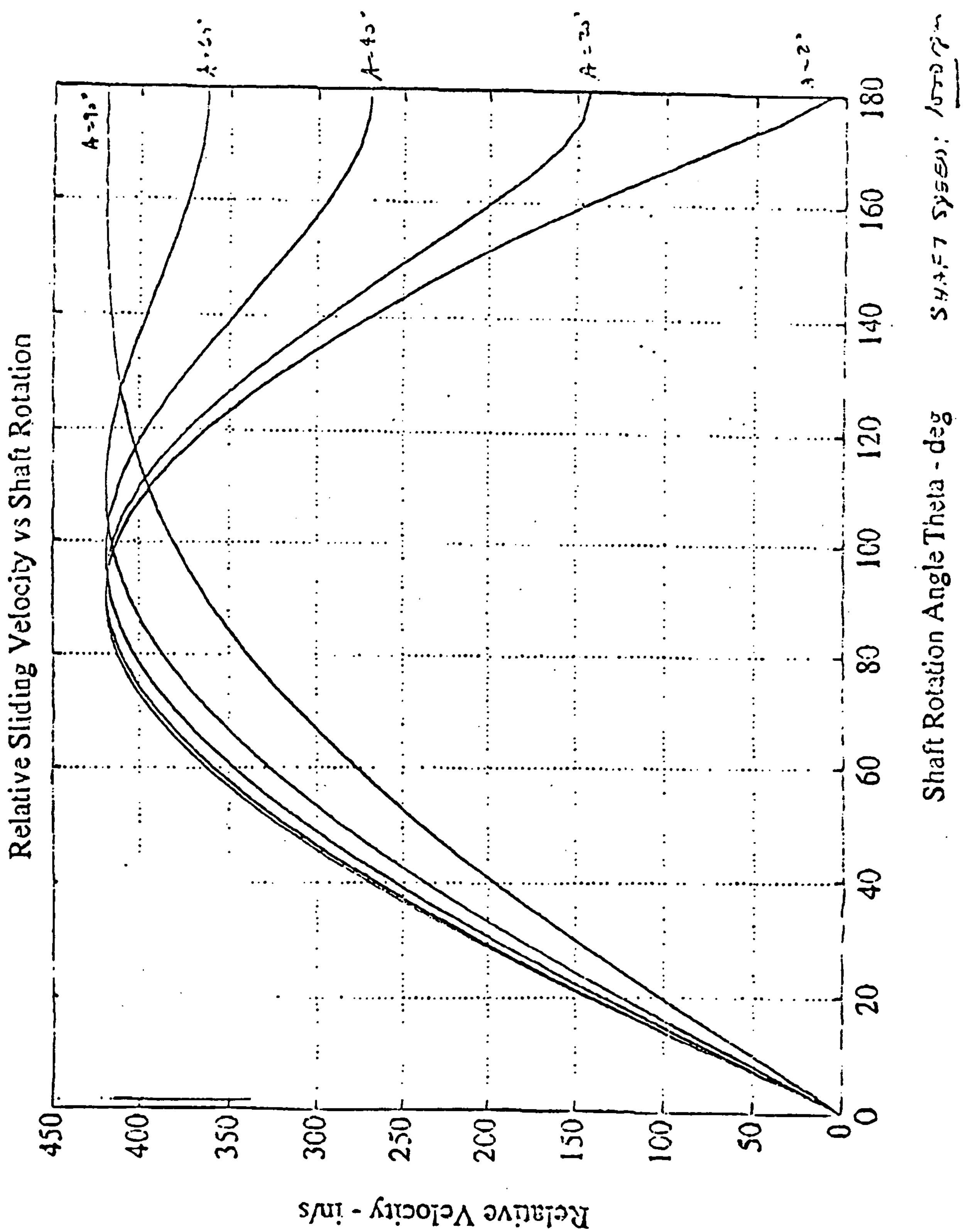


FIG. 19C

FIG. 20


Relative Sliding Velocity vs Shaft Rotation
Relative Velocity - in/s
Shaft Rotation Angle Theta - deg
0
50
100
150
200
250
300
350
400
450
20
40
60
80
100
120
140
160
180

INTERNATIONAL SEARCH REPORT

International application No.
PCT/US99/11642

A. CLASSIFICATION OF SUBJECT MATTER

IPC(6) :F01C 3/08
US CL :418/190, 195; 29/888.023
According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

U.S. : 418/190, 195; 29/888.023

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	US 0,032,372 A (JONES et al.) 21 May 1861 (21.05.61), page 1, col. 2, lines 35-48.	1
X	US 1,379,653 A (SHOEMAKER) 31 May 1921 (31.05.21), page 1, lines 98-107, figure 3.	1
X	JP 43-29764 B (KOMURO) 20 December 1943 (20.12.43), abstract and figures 1 and 2.	1
X	DE 2364281 A (SCHUKEY) 26 June 1975 (26.06.75), abstract and figure 1.	1
X	DE 3221994 A (WERNER) 15 December 1983 (15.12.83), abstract and figure 24.	1
X	US 5,039,289 A (EIERMANN et al.) 13 August 1991 (13.08.91), col. 4, lines 47-55.	8

[X] Further documents are listed in the continuation of Box C. [ ] See patent family annex.

* Special categories of cited documents:

"A" document defining the general state of the art which is not considered to be of particular relevance

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"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)

"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art

"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
28 JULY 1999	02 SEP 1999

Name and mailing address of the ISA/US	Authorized officer
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